A COMPARISON OF FORCED AND NATURAL CIRCULATION STEAM GENERATORS

W. L. MARSHALL

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By

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Lieutenant Commander, United States Navy

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PREFACE

For a number of years, especially since the end of World War II, the engineering literature has contained an increasing number of references to forced circulation boilers. Most of these articles extol the virtues of some one kind of boiler or tend to justify the conventional natural circulation principles. The range of boiler types covered by the term "forced circulation" is much broader than usually realized, and since some of these types are basically quite different the existing confusion on this subject is understandable.

We commonly think of the United States as a leader in technological progress and development, and in many fields that is true. There are other fields, however, where from time to time we tend to lag. This has been especially so with regards to the application of forced circulation in boilers even though some of the working principles were originally discovered in this country. It is gratifying to note the rapid and accelerating interest which is now being shown towards making up the lost time.

In this paper an attempt is made to classify and describe the common types of forced circulation boilers. By comparison with natural circulation principles, advantages and disadvantages are brought out. The comparison is not intended to minimize the importance of natural circulation, but rather to show that forced circulation represents a step forward in steam generation similar to that of the watertube over the firetube boiler.

I wish to express my appreciation to Professor E. E. Drucker for his guidance, assistance, and willing cooperation in helping to acquire

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SUMMARY

In this paper the various types of forced circulation boilers are described along with the motivating factors both as regards to design and economics which have stimulated their development. In order to show the inherent advantages and limitations, a series of circulation calculations for a natural circulation boiler are made and the associated problems discussed. The effect of applying forced circulation to this situation is then described. Other changes in the boiler characteristics resulting from the application of this principle, such as the use of small tubes, are discussed and illustrated with calculations where applicable.

Early and interesting marine installations utilizing various types of forced circulation are described. Also included are comparative data based on several years of operation of a forced and a natural circulation boiler operating side by side in an eastern industrial plant.

NOMENCLATURE

Top dryness (X)	- Percent by weight of steam at the top of a generating tube.
Circulation ratio	- Reciprocal of top dryness or ratio of the mass rate of water to the mass rate of steam.
Controlled circulation	Design features which permit the proportioning of the water supply to a boiler section or individual tube. This is accomplished by means of orifices at the inlet of the steam generating circuits.
Downcomer	- A tube designed to carry water from the steam drum down to the mud drum or lower boiler header.
Riser	- A steam generating tube.
Bifurcated tube	- Two small tubes which unite into a common tube prior to joining a header or drum.

f - Dimensionless friction factor used in the Fanning equation.

u - Velocity with appropriate subscripts in feet per second.

z - Tube height in feet $= z_2 - z_1 = z_o - z_1$

D - Diameter in feet.

g - 32.2 feet per second per second

$\dot{m}$ - Mass rate of flow in pounds per hour

V - Specific volume in cubic feet.

d - Density with appropriate subscripts in pounds per cubic foot.

A - Area in square feet

k - Thermal conductivity, BTU per hour ft. degree F.

v - Viscosity, pounds per hour feet.

c_p - Specific heat, BTU per pound degree F.

h_i - Heat transfer coefficient (inside) BTUs per hour square foot deg. F.

q - Rate of heat absorption BTUs per square foot per hour.

t - Temperature in degrees Fahrenheit.

r - Radius in feet

σ - Stress, pounds per square inch.

a - Inside tube radius in inches.

b - Outside tube radius in inches

E - Modulus of Elasticity, psi

α - Thermal Coefficient

ν - Poissons ratio, 0.3 for steel

u - Radial displacement

ϵ - Strain

p - Pressure

I. INTRODUCTION TO FORCED CIRCULATION

1. Brief Historical Background and Comments

Novel or special boilers are generally considered to be an outgrowth and result of the design advances of recent years. Surprisingly enough this is usually not so, and particularly is this true of forced circulation boiler designs.

As far back as 1828 Jacobs Perkins designed and built a high pressure series, or coil, boiler, operating on what we now call the "once through" principle to be explained later. Only the lack of suitable measuring instruments prevented him from proving the great step forward his boiler represented. Both his boiler and his ideas were far ahead of his time, and a conservative engineering world failed to take advantage of his genius. In 1877 and 1879 Herreshoff, and in 1880 Serpollet demonstrated that such designs were successful and could give good efficiency and service. About twenty years later, in 1897, De Laval stimulated the engineering world with his high pressure (2800 psi) coil boiler and turbine at the Stockholm Exposition. All of these were considered novelties in their day, since natural circulation was more than adequate for the boilers then in use. Boiler making was considered an art rather than a science and this engendered conservatism which when coupled with the lack of knowledge of what went on inside tubes and furnaces successfully prevented any serious development of the "once through" forced circulation principle.

In 1856, Martin Benson of Cincinnati, Ohio, invented the operating principle of the forced recirculation boiler which is the other main

category of forced circulation boilers. By 1859 approximately forty-five such boilers were operating in the United States. Pressures and temperatures in use at that time were low, and while Benson's boilers were successful, no particular advantage was attributed to this use of forced circulation. Consequently after the initial flurry of construction, no further use was made of this principle for many years.

Beginning about 1923 Benson's ideas were revived by W. D. LaMont in what is now known as the LaMont type boiler. This revival was brought about by the coal shortage which followed the exploitation of European coal fields during World War I. The need for greater efficiency was imperative. This coupled with the conception then prevalent, but since discredited, that conventional natural-circulation boilers would be unable to operate at pressures much above 800 psi stimulated the development of forced circulation boilers of all types, particularly the LaMont design.

The demand for increased economy with its attendant higher pressures and temperatures created and is still creating problems in the metallurgical field. The introduction of new alloy steels in recent years permits operation at temperatures and pressures unheard of twenty-five or thirty years ago. As still higher temperatures and pressures are attempted, an apparent temperature limitation for alloy steels is being reached. In an attempt to overcome this limitation, the metallurgists are currently experimenting with ceramic coatings to protect the metal and allow temperatures still higher than the current tentative limit of 1150°F.

All modern steam generating units employ the forced circulation principle in all parts except the water cooled furnace and boiler; the air heater operates under forced circulation due to forced- and induced draft fans; the economizer employs forced-circulation by virtue of the feedwater pump; the superheater also operates under forced circulation because of the large pressure drop between the boiler drum and the superheater outlet. In each of these pieces of apparatus proper distribution of heated and heating fluid depends on the pressure drop across the apparatus which represents the degree to which forced circulation is used.

Modern watercooled furnaces and boilers employ many more parallel circuits to handle the heated fluid than are to be found in the auxiliary apparatus above. One of the major design problems is the distribution of water to each circuit and the providing of adequate circulation in the right direction. In natural circulation the designer does not have full control over this problem, because variation in heat transfer causes variation in circulation. The circulation is started and maintained by the heat absorption into the water and steam and a miscalculation as to the intensity of heat absorption may cause circulation troubles. The rate of firing and circuit location in the furnace cause wide variations in circuit velocity and steam-water ratio.

From the design viewpoint it seems desirable then to apply forced circulation to the only parts of the steam boiler where it has not been generally used, namely, the boiler and furnace.

2. Forced Recirculation Principle

Natural circulation boilers are composed of two main types, straight-tube or bent tube. The circulation problems are essentially the same in each. Forced circulation boilers are also of two main types, viz., (1) forced recirculation, and (2) forced flow, once through. An external feed pump is used with both types, the principal difference between the two, from the standpoint of water circulation, being in the amount of water which is pumped through the boiler circuits.

In the forced recirculation type the amount of water supplied to the boiler varies from one hundred and ten percent to as much as twenty times the amount evaporated. Evaporation is not carried beyond the point where the dissolved solids can be retained in solution in the concentrated boiler water at the heat transfer surface. The mixture of steam and water is then discharged into the drum where the steam is separated from the water and then through the superheater. The excess water passes through the boiler where its temperature becomes saturation temperature corresponding to boiler pressure. It flows then through a heat exchanger where it gives up some of its heat to feed water and flows thence to the feedtank. The feed pump takes suction from the feedtank, recirculating the excess water which accumulates there, back through the boiler circuits. The concentration of solids is eliminated as in conventional boilers by a periodic blowing down of the steam drum. This type of boiler can employ the conventional type of controls, since feedwater rate is regulated to maintain a water level in the drum, and the regulation of fuel and air may be

either manual or automatic, as in a conventional natural-circulation boiler. However, since some water is recirculated, one or more circulating pumps are required.

In a variation of this forced recirculation type, the evaporation of water is accomplished by superheated steam in a drum outside the boiler setting which eliminates the hazards from solids being deposited on the heating surface. The evaporated steam is pumped through the superheater. That portion of superheated steam not delivered to the prime mover is returned to the drum to evaporate feed water by direct contact. This type of unit has a steam-circulating pump, in addition to the feedwater pump.

3. Once Through Principle

The once through type of boiler is quite different in many respects from the forced recirculation type. No excess water is pumped through the system, hence the designation "once through". The feed water enters the tubes, is heated, then flashes into steam, and is superheated all in one pass through the boiler. Any dissolved salts or solids from the feed water are deposited on the tube surfaces as evaporation takes place. Because of the way in which the water flashes into steam in the hot tubes, this type of boiler is sometimes referred to as a "flash boiler". Since these boilers usually do not have a drum the storage capacity is very small and the boiler controls must be quick-acting and reliable in order to handle sudden load fluctuations.

The biggest problem with these boilers has been the elimination of the salt and solid scale deposits in the tubes. Since this scale has high heat resistance, unless it is periodically eliminated, tube failures result. Various methods have been developed for washing these deposits out of the tubes. In the Benson boiler, for instance, one group of tubes at a time is isolated from the others, by special valves and piping connected to their inlet and outlet ends, feed water is forced through the tubes, washing out all the salt deposits, and the tubes are then reconnected into the boiler circuits. The water used in this process becomes heated in its passage through the tubes so it is usually passed through a feed water heater where it gives up most of its heat and then since it is contaminated with salts it is thrown away. In another type of boiler, a salt interceptor is installed at a point in the circuit where the steam still carries five to ten per cent highly contaminated moisture; the contamination resulting from the concentration of most of the salts originally contained in the feed water. The interceptor is periodically blown down to get rid of the accumulation. Make up feed is supplied equivalent to that blown down. In all these once through boilers feed water is circulated through the circuits as the boiler cools down while being secured in order to prevent overheating and to flush out accumulated salts in the tubes.

II. DESCRIPTION OF VARIOUS TYPES OF FORCED CIRCULATION STEAM GENERATORS

1. Introduction

Aside from two large units which will be discussed later and a few experimental types, forced circulation boilers are practically unknown in this country. The current trend to higher temperatures and pressures coupled with an increasing need for economy and efficiency are causing a mounting awareness of the advantages of forced circulation.

The recirculation types of forced circulation boilers are the ones which are receiving the most attention in this country. There are five main types in this category, namely: LaMont, Loeffler, Steamotive, Velox, and Schmidt-Hartmann. Each of these will be discussed individually.

The "once through" principle is best illustrated by the Benson, Ramzin, and Sulzer types. These boilers are most popular in continental Europe, especially Germany, Russia, France, and Switzerland. Little attention has been given to them in this country, but since they have certain inherent advantages they will be described in detail.

2. Forced Recirculation Types

A. La Mont

The first LaMont type steam generator was the one developed by Martin Benson in 1856. (See Figure 1) His boiler consisted of a number of tube coils connected to a supply header at their lower or intake end. These had "choking plugs" or orifices inserted in the branch tubes to regulate the flow of water into each tube coil. This gave control of the water distribution to each tube. The supply header was fed by means of

a water circulating pump, the suction side being connected to the water space of the steam and water drum placed near the boiler and outside the boiler setting. The upper or steaming ends of the tube coils were connected to a collecting header through which the water-steam mixture was discharged into the steam space of the water and steam drum previously mentioned. Contemporary test reports * show that the optimum operating conditions were achieved by circulating six to eight times as much water through the boiler as was evaporated, and the pressure drop through the distributing orifices was two to three pounds per square inch; the pressure to be overcome by the circulating pump is stated to have been seven to ten pounds per square inch. The diameter of the tubes constituting the evaporating coils seems to have varied from one to one and one-half inches in the numerous boilers built. The starting time is claimed as having been only six to twenty minutes. Benson recognized the necessity of adequate steam space in the drum, and recommended one cubic foot of steam space for every twenty-two square feet of heating surface in boilers of two hundred to one thousand square feet heating surface; one cubic foot per 25 square feet in boilers with over one thousand square feet heating surface; and one cubic foot per twenty-seven square feet in marine boilers.

* J. F. Spencer: "Description of New Construction of High Pressure Steam Boiler", Proc. Inst. Mech. Eng., 1859, p. 264

J. J. Russell: "On Benson's High Pressure Steam Boiler", Proc. Inst. Mech. Eng., 1861, p. 30

In addition to the forty-five boilers of this type built in this country, the success of Benson's innovation is further attested by the fact that Messrs. Hawthorn, using Benson's British Patent (1858), constructed a marine boiler in England which was extensively used for tests. The boiler had one inch inside diameter tubes. It was a three hundred forty square foot boiler with nine square feet grate surface. One especially interesting test consisted in the continuous operation of the boiler at eighty pounds per square inch pressure for fourteen days with a fifteen per cent sea salt solution in order to ascertain the serviceability of Benson's design for marine use. Examination showed the lower tubes to contain an internal scale one-sixteenth inch thick while the upper tubes were practically scale free.

The success of the LaMont boiler is due quite largely to its comparative simplicity of construction and also to the fact that its design is nearer to the conventional boiler than any other forced circulation type.

Figure 3 gives the flow cycle and general schematic arrangement of tube surface. Boiler water passes from the drum which is for all practical purposes outside the boiler proper, to the suction side of a circulating pump. This pump must overcome the friction and orifice drops. Orifices serve the purpose of distributing the feed water in accordance with the heat absorption of each tube. In most of these boilers the water circulation is adjusted in each tube by the size of the orifice so that at normal boiler load each tube receives about four or five times as much water as the quantity of steam generated in the tube.

This makes overheating of the tubes impossible, because high velocities of the steam and water are assured at all loads.

From the tubes the water and steam mixture is discharged into the boiler drum, where they are separated. The steam is drawn off to the superheater while the feed pump supplies an equivalent quantity of water to the water circuit. Feed water control is maintained by any standard type of feed water regulator. The pressure drop which must be overcome by the pump is the hydraulic friction of the circuit and will amount to about 30 - 40 psi. This is in the neighborhood of 0.5 per cent of the boiler energy output. These pumps, however, are usually turbine driven and care must be used to utilize the exhaust to its best advantage, else the pump loss becomes considerably higher. Electric drive is usually provided for starting these pumps or for normal use with steam as a standby.

An adaptation of this LaMont principle which has proved especially popular in Germany and Sweden is shown in Figure 6. This shows the addition of a forced circulation LaMont water wall in an old natural circulation boiler in order to increase its output. It has the advantage of cheapness and convenience. Especially advantageous is the feature that the water wall tubes do not necessarily have to be in the perpendicular and that by using coiled or inclined tubes, one vertical distributing and collecting header each is sufficient for four water walls as shown in Figure 6. If only one circulating pump is provided per boiler unit, provision can be made to maintain some amount of positive flow through the boiler by automatically connecting the feed water

supply to a point in the circuit, so that feed water is discharged into the distributing headers of the circuit. The water walls of the LaMont type have proved exceedingly useful where addition of a water wall has proved necessary in natural circulation boilers when the addition of natural circulation water tubes would be impossible because of lack of accommodation and access. The addition of water wall surface may be considerable even in small boilers. In the installation reproduced in Figure 6 the water wall contributes 24,200 pounds of steam per hour to a total evaporation of 44,000 pounds per hour.

Since the boiler drum is retained in LaMont boilers the storage capacity is sufficient to handle sudden load fluctuations, an important advantage. It is, therefore, unnecessary to have sensitive and quick-acting control mechanisms as in boilers of the once-through type which have no storage capacity.

An important operating advantage is that water circulation can be established before and while starting the boiler. This allows a gradual heating up of the whole boiler circuit which serves to minimize expansion stresses. Starting time is consequently very short.

Another important application of the LaMont boiler is for waste heat recovery from the exhaust gases of marine diesel engines. The special design developed for this use is shown in Figure 7. In this design the diameter is about two and one-half times, and the length four to six times, the outside diameter of the exhaust pipe of the engine. The amount of waste heat extracted will, of course, vary with the engine employed, but, in general, for an average four-stroke

Diesel main engine which exhausts 180 cubic feet of gas at 850° - 900° per H.P.-hour, if the final temperature is to be lowered to 400° F, about 1300 BTUs per H. P-hour can be recovered.

It is perhaps interesting to note in passing the great similarity between modern LaMont type boilers and those of Martin Benson of nearly one hundred years ago.

LaMont boilers and water walls have been utilized by most of the European countries for many years, especially in Germany and Sweden. Between 1932 and 1946 in Sweden alone more than three hundred fifty of these units were constructed. At first this construction consisted mainly of the modification of existing boilers by the addition of LaMont water walls which in many cases doubled the output of the boiler. Later the saving in weight and space made possible with this type boiler caused the always thrifty Swedes to turn more and more to the manufacture of LaMont boilers in preference to natural circulation designs. An interesting feature of these Swedish boilers is their versatility of firing. Since native coal is scarce in Sweden, and since coal imports were negligible during World War II, these boilers were built or adapted to burn almost anything. Some were built to burn pulverized coal, oil, gas, or wood, and most are capable of burning three kinds of fuel. In Germany, Britain, and other countries the LaMont design is utilized on a considerable scale where space and weight are important factors, but not to such an exclusive degree as in Sweden. By 1941 there were well over seven hundred LaMont units operating in Europe and the figure today is much higher.

This same year, 1941, saw the first commercial application of forced circulation in this country. This LaMont type boiler, conforming to U. S. specifications, was built by the Combustion Engineering Company and installed in the Montaup Electric Company plant in Somerset, Massachusetts. It has proved highly successful. Development of forced circulation was curtailed by the war years so that only one other sizable installation was made during the War and that was in 1943. However, within the last two years an increasing number of these boilers have been ordered.

B. Loeffler

This unique boiler was developed by an Austrian engineer, Dr. Stephan Loeffler, and was first manufactured commercially by the Iron and Steel Works, Vitkovice, Czechoslovakia, in the early 1930s. The only water in the heat absorbing surfaces is that contained in the economizer tubes. Steam is employed directly as the heat carrying medium, thus completely eliminating the possibility of scale formation in the boiler tubing which is the underlying motive of this unorthodox design.

Steam moving in forced flow through the boiler circuit, is alternately superheated in the heat absorbing, or boiler, surface and then desuperheated in direct contact with water. The amount of steam generated by the desuperheating process is made identical with that of the steam tapped off at point of maximum steam temperature, which coincides with the outlet of the boiler surface. Figure 8 illustrates this rather simple arrangement. In this figure (S) represents radiant

part of boiler furnace, (N) the convection portion, (B) the evaporator, (U) the steam circulation pump, (V) the economizer, (L) the air pre-heater, (P) the feedwater pump, (D) the steam tapped off for use.

All steam is generated in vessels not directly exposed to furnace heat, the tubes containing only the dense dry steam at pressures 1500 psi and above, usually about 1900 psi and 900°F. At this high pressure the specific volume is low, so the density is high. If pressures below about 1500 psi are used the circulating pump dimensions become unduly large because of the increased specific volume of the steam and the pump power requirements become too great. Prior to World War II efforts were being made to reduce the steam consumption of the circulating pump to make a pressure of 750 psi practicable. These efforts were resumed after the War, but the progress, if any, which has been made is not known. It is the low specific weight (approximately 0.21 pounds per cubic foot) that makes it economically possible to circulate the steam by means of a pump, and the velocity of circulation is such as to operate with small temperature gradient between the steam and metal wall. The steam velocity approaches 65 feet per second and a heat transmission rate of 40,000 B.T.U. per hour per square foot per degree Fahrenheit is realized without overheating the tubes. The main value, however, lies in the fact that water purity is of secondary importance, since no scale forms on the wall surfaces. All scale forming substances collect in the evaporator which is external to the boiler furnace and where these substances may be blown down at will.

The pump is usually of the single impeller type rotating within a housing. The main purpose of the pump is to circulate the steam through

the tube system at such a rate that the final temperature remains nearly constant at all loads. A known quantity of heat may be transmitted through the tube walls and imparted to a large quantity of steam at low tube wall temperature or to a small quantity at high temperature. Speed control of the pump is therefore essential. To vary the boiler output simple combustion control can be used.

Since these boilers suffer no detrimental effects from poor quality feed water, they have been used in continental Europe, and England where local water conditions present a problem and have been highly successful.

C. Steamotive

The steamotive boiler is a development of the Babcock and Wilcox Company assisted by the General Electric Company and the Bailey Meter Company. Its original design was for use in a steam locomotive so that compactness, simplicity, flexibility and range of operation, automatic controls, and possibility of burning a wide range of fuels are characteristic of this design. Outstanding design features are the forced draught pressure combustion at about 20 in. water pressure in the water-cooled furnace and the forced flow once-through principle combined with re-circulation at all loads of an excess water quantity equivalent to ten per cent of the maximum boiler output.

The flow diagrams are shown in Figure 5. From the burners the flame and gases pass horizontally through the completely water-cooled furnace, then up and back with a one hundred eighty degree turn into the superheater, around the separator, and through the economizer and air heater. Air for combustion being under pressure forces the air and

resulting gas through the whole air-gas circuit, no induced-draft fan being used.

The feed water after leaving the economizer divides into five circuits as shown, forming the floor, sides, roof, and two loops of the furnace screen. All steam is formed in the furnace and with the residual water passes to the separator. As in the Velox boiler the inlet connections of the boiler circuits to the separator are arranged tangentially, and the tube ends flattened to create a jet effect. This sets up a whirling motion of the steam-water mixture which has proved very effective in separating the two phases. The remaining water contains a high salt content resulting from the concentration of the salt contents in the feed water during the process of evaporation into the remaining water. This excess water carrying the impurities, and termed the "spill-over", is drained from the separator, passes through a heat exchanger to a hot well where it is refed to the boiler. The steam leaves the separator and passes through the superheater under temperature control and then to the turbine. A reciprocating type of feed pump is used with the combustion air blower mounted on the same shaft. A "fixed resistance" tube similar to an orifice maintains "spill over" constant irrespective of load while various controls maintain the separator drum water level constant and prevent undue rises in water level in the case of sudden load drops.

Following the above design an improved Steamotive unit (Figure 9) was developed for small, land, power-plants, which has also been adapted to experimental marine work in the past several years. This is a vertically fired unit with a circular furnace and two intermediate open passes

between furnace outlet and superheater. Both furnace and gas passes are completely watercooled. The open gas pass arrangement provides increased flame length and ensures complete combustion prior to the entering of the furnace gases into the superheater.

All units so far built are equipped for oil firing and considerably more research will be needed to adopt this boiler to pulverized coal firing.

It is perhaps of interest to note that the earliest experimental boiler of this type was completed in 1926 and was tested at steam pressures from 1500 psi to 5000 psi at a capacity limited only by that of the pump. This capacity was about 5000 pounds per hour corresponding to a mass flow of about 475,000 pounds of steam per hour per square foot area of tube, with a final steam temperature of the order of 800 degrees Fahrenheit. The boiler consisted of seven circuits of one inch tubes with one-half inch inside diameter.

D. Velox

This steam boiler was first suggested in 1927 by the Research Department of the Brown-Bovari Company as a method of utilizing the large heat losses encountered in the experiments connected with the development of the gas turbines of the Holzwarth type. In January 1928 followed the first patent applications and shortly thereafter a small test boiler was constructed, upon which the original test measurements of heat transfer were conducted. During this same year a large ten-ton trial boiler utilizing oil fuel and a smaller unit for gas were constructed. These first two boilers were operated on the explosion principle and a combustion chamber was essential. Although the small unit operated quite

satisfactorily, difficulty was experienced with the ten-ton unit. The large combustion chamber (over one cubic meter) failed to give uniform explosions and the pressure expected. Considerable research and development including the application of pressure firing continued until in 1931 there arose out of the reconstructed explosion boiler a steam generator operating with constant pressure, which has since been given, in recognition of the principle employed, the name "Velox".

The principle of operation of these boilers is quite ingenious. From Figure 10, it can be seen that air under pressure enters the furnace at a high velocity. The resulting high flame velocity and consequent very large heat transmission requires a highly efficient evaporating system. This is provided in the form of a circular furnace lining composed of a number of evaporator tubes through which water and steam are forced by a special circulating pump. These cylindrical, vertical evaporator elements are equipped with internal flues into which the gases enter at the bottom of the evaporator element as shown in Figure 11. Due to the high gas velocities only about twenty-five per cent of the total heat transmission in the unit is effected by radiation, in contrast with orthodox boiler designs which absorb as much as fifty percent by radiation. Figure 12 shows an evaporator element equipped with an integral superheater element placed in the flue space. This is now a common feature of Velox boilers. The flue gases leave the top of the evaporator elements and pass to a gas turbine which in turn drives an axial compressor, that in turn provides the high velocity pressurized air for the combustion. The exhaust from the gas turbine in turn is discharged into the economizer where additional heat is

extracted. The steam and water from the evaporator tubes passes over to a steam separator drum where this mixture is introduced tangentially. The resulting whirling motion in the drum results in a steam releasing capacity which is one hundred - two hundred times greater than could be obtained from a horizontal drum-water-level. The steam discharge is quite dry and the danger of priming is obviated. After separation the steam flows to the superheating elements and thence to the turbine. The lower part of the drum which collects the centrifuged water acts as a mud drum.

Since the Velox boiler is not self-starting, an auxiliary electric motor or steam turbine must be provided for this purpose.

As previously mentioned the gas velocities in this boiler are high. This stems from the original research of the Brown-Bovari Company in which it was found that at velocities in excess of the velocity of sound, a heat loss is incurred in a gas turbine nozzle far in excess of that computed theoretically by the extrapolation of Nusselt's formula for the heat transfer from gases flowing through tubes. The Velox steam generator is an outgrowth of this observed phenomenon. In a typical boiler such as Figure 10, the combustion chamber is lined with vertical tubes, about four inches in diameter, which contain the water. Each of these tubes contains three one inch diameter tubes which channel the flue gases at velocities from 660 to nearly nine hundred feet per second. It is in these tubes that the enormous heat absorption rates obtain. The radiant heat of the furnace is also effective on the furnace side of the large tubes surrounding the furnace. At the outlet of the combustion chamber

the gas temperature is about 3200°F. at about 36 psia. At the outlet of the small tubes this temperature has been reduced to about 1500°F. and a pressure of 33 psia. The gases then enter the superheater (5) and are reduced to around 900°F and 31 psia. From the superheater gas outlet the flue gases pass to the gas turbine (6) where the temperature drops to 700°F. and the pressure to about 16.5 psia. From the turbine the gases are forced through the economizer (7) at a velocity of about 400 feet per second and discharging the gas at about 200°F. At the same time this exhaust gas turbine drives the air compressor which compresses the air to the combustion chamber to 36 psia and 300°F.

The first commercial Velox boiler was built in 1932 for the Mondville Steel Works in France. In 1933 seven were ordered, this figure rising to 31 units in 1935, and to 75 units in 1940. A comparatively large number of these machines have been built during the last twelve years.

Although this boiler has proved highly popular abroad, it has received only scant attention in this country. This is in part due to several limitations. At present its application is limited to the burning of gaseous and liquid fuels and in size it is limited to maximum outputs of approximately 200,000 pounds of steam per hour. The limiting factor is the exhaust gas turbine which in a 200,000 pounds per hour unit must have an output of about 6,000 B.H.P. Because of the high temperatures in the gas turbine, it has proven difficult to increase the dimensions of the turbine rotor. Some success has been obtained in the past two or three years in overcoming this obstacle.

Considerable work has been done towards utilizing pulverized coal but it has proven difficult to overcome the harmful effects of ash in the flue gas on the evaporator tubes, nozzles, and gas turbine. Since the main boiler customers in the United States are the central power plants which are predominately coal fired and concerned with large outputs the two limitations mentioned above preclude the early appearance of this boiler in the United States unless some special application is found.

In a Velox unit the efficiency of the gas turbine will run about 83 per cent and that of the blower 73 per cent, so that the overall efficiency is 60 per cent. This is not realistic, however, because the losses in the compressor except for bearing and casing losses remains in the cycle as increased air temperature. Consequently the major portion of the power required for compressor work remains in the system and does not affect the efficiency of the unit.

Figure 10 is a Velox boiler with a separate superheater. This type is used where poor quality oil tends to cause clogging of gas passages in built-in superheaters. It is becoming more popular than the integral superheater since poorer grade oils may be used as fuel.

E. Schmidt - Hartmann

The Schmidt-Hartmann boiler is the best known of the so-called "indirect steam generators" in which the boiler water does not actually come in contact with the hot furnace gases. It has in reality two circuits as can be seen in Figure 4. It may be either natural circulation or forced circulation depending on the operating pressure. It is of the recirculation type where water from the separator drum is kept

separate from the feed water and recirculated. The steam which has been separated from the water in the separator and which is free of salts, is carried to the main drums where it gives up its heat in passing through a coil bundle submerged in the feed water. The steam evolved from the feed water is superheated in the conventional manner and delivered to the turbines.

The primary heat carrier, or separator, circuit must operate at a higher pressure than that of the secondary, or feed, circuit in order to obtain the temperature margin required for the heat transfer from the condensing circuit to the water evaporated in the main drums. Since this temperature margin is expressed by the difference in saturation temperatures of heating steam and feed steam, it would at first seem as if undue pressure differences would be required in order to create a sufficiently high temperature difference. However, due to the excellent over-all heat transfer coefficient between condensing steam and secondary feed water in the main drum, the required temperature difference is much smaller than would be presupposed.

As Figure 4 shows, the separation of the water-steam mixture in the primary circuit occurs in the separator, the steam flowing to the main drums' tube bundle and the residual water is passed into a small header or to the distributing drum. These elements and also the main drums are located outside the furnace area and are not exposed to the boiler heat. They are, therefore, free of the heat stresses due to flame exposure.

The primary, or heating, cycle is a closed affair which experiences very little water loss due to leakage. What little occurs is made up with condensate at periodic intervals. This boiler has ample storage

capacity so that there will always be enough time available to shut the boiler down in order to prevent drying up and burning of the boiler tubes in cases where excessive water loss occurs by some accident, such as the bursting of a boiler tube or heavy leakage of joints. The condition of the primary cycle is indicated by a combined pressure and temperature gauge which indicates whether the primary steam is saturated or superheated. If superheat is indicated then the cycle is short of water; and the seriousness of the shortage is indicated by the degree of superheat shown.

Because of the small head of water over the tube bundle in the main drums and the lack of water and steam spouts from the risers as in a conventional steam drum, the evaporation which takes place in this boiler is unusually quiet and subdued. As a result there is much less moisture carry over even with heavy salt concentrations. Even at heavy loads the moisture content in the saturated steam will not exceed 0.01 per cent. This is many times better than in conventional boilers and means that moisture and salt carry over to the turbines is a minimum. Tests have shown that boiler water concentrations as high as 20,000 - 30,000 parts per million can be admitted in the evaporator drum without endangering the superheater or the turbine.

3. Forced Flow, "Once Through" Types

A. Benson

In 1922 Mark Benson conceived the idea of generating steam in a once-through boiler at critical pressure where no steam bubbling would occur. It was believed, then, that steam generation in small tubes at sub-critical pressures would meet with serious difficulties due to unequal

distribution of the working medium and steam bubbles adhering to tube walls, taking up too much of the tube area, and therefore interfering with heat transmission from the tube wall to the water. With critical pressure, Benson hoped to avoid the tube failure troubles of earlier once-through boilers since at this pressure no volume increase in transforming water into steam can take place. His design seemed most promising because it made available steam at any subcritical pressure by throttling and then securing any desired initial steam temperature by reheating. An experimental unit was built by the English Electric Company, Ltd., at Rugby, England which proved the feasibility of boiler operation at critical pressure.

These prospects coupled with the influence of the coal shortage in Germany induced the Siemens-Schuckertwerke of Berlin in 1924 to undertake commercial development of Mark Benson's idea. But the first Benson boilers developed tube failures like their subcritically operated predecessors. The non-existence of actual evaporation due to critical pressure operation overthrew all theories on sticking steam bubbles and proper distribution of flow could be checked by the temperature of the working medium at the outlet of the tubes and elements.

The greatest practical difficulty encountered, which took several years to discover, lay in the fact that in the region where water transforms into steam serious salt deposits occurred in the tubes. These deposits due to their poor heat conductivity, were responsible for a considerable increase in tube-wall temperature which in turn made the tubes liable to failure if the deposit zone was located in the radiant portion of the heating surface. This was the case in the earliest Benson boilers.

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The reason it took so long to discover this fundamental cause of tube failure lies in the circumstance that whenever a tube failed, the actual deposits could never be found, for, being water-soluble, they were washed away by the steam and water mixture escaping through the crack before the boiler could be stopped.

As a result of this discovery the deposit zone was removed from the combustion chamber, or radiant zone, and located in the convection zone where the gas temperatures are low enough not to endanger the strength of the tube walls. This automatically involved spreading of a given amount of deposits over a greater tubular surface because of the lower heat intensity in the convection-heated portion. This in turn meant formation of thinner deposit layers.

A second conclusion was that the tube failures of earlier once-through boilers were probably due to the same kind of salt deposits. It was, therefore, deemed possible to operate the Benson boiler at sub-critical pressures as long as the deposit region was kept in the convection zone. This proved to be true and since then all Benson boilers have been operated at suitable subcritical pressures. The salt deposit rate and weight of the pressure parts both decrease with reduction in pressure which makes for improved operation and lower initial cost.

The salt deposits are disposed of when the boiler is stopped by flushing it for a while with feedwater to prevent overheating of the otherwise dry tubes due to heat from the brickwork and baffles. Benson boilers have been operated continuously in this way for periods up to

two months. For longer periods of uninterrupted service a simple scavanging device is provided which allows flushing of different groups of the salt deposit zone progressively while the boiler is under load.

Equal flow distribution over all tubes connected in parallel is assured by the arrangement of mixing headers. These headers are required anyway from the construction point of view since it is easier to collect water in one interconnecting pipe than to transfer it in twenty or more parallel tubes when passing from say the economizer to the radiant part of the furnace.

The Benson boiler has no drums, no recirculating pumps, an unlimited fuel range, a salt deposit zone located outside the furnace proper, is available for pressures up to critical, and does not need outside steam for starting and like all forced boilers is lighter and smaller and consequently cheaper.

The flow diagram of a Benson boiler is reproduced in Figure 13. The water from the feed pump is passed through an economizer (a) which is arranged for counterflow between water and flue gas. The outlet is connected to the waterwall inlets. As Figure 13 shows, the water tubes are combined into groups through which the water flows in succession. In order to obtain an upward flow in the heating tubes the economizer outlet is connected by an unheated tube with the bottom header of the second group (b). The outlet of this group is connected to the bottom of a third group (c) and a similar mode of connection is used to the last group (d) which is the superheater group. This superheater is

likewise arranged for parallel flow in order to avoid excessive heating of the superheater tubes.

As in the case of all once-through boilers, reliable quick-acting, yet not too complicated, controls enabling the boiler to follow the most rapid load fluctuations without delay are essential. It has been found that, in general, boiler control is best effected by deriving the control impulses from the steam temperature instead of from the boiler pressure. Actual experiences, however, have shown that the Benson boiler can easily be made to operate at varying pressure, with the variation in boiler pressure being a function of the load. This and the fact that the specific steam volume and steam pressure decrease approximately proportionally, if steam temperature remains constant, have given rise to the suggestion of regulating the turbine power output by pressure changes at the turbine inlet, without use of a turbine throttle. In this case the heat drop and volume of steam passing through the turbine will remain nearly constant over a wide load range which in turn results in a substantially constant themodynamic efficiency and constant specific heat consumption over the load range. Such a variable pressure Benson boiler was in operation in the late 1930s in Berlin in connection with a radial turbine of the Ljungstrom type, the steam pressure varying from 426 to 1704 psi.

In 1946 Mark Benson proposed a modification of the standard Benson boiler which, it is claimed, eliminates the feed-water difficulties common to "once through" boilers. The essential feature is the addition of an evaporator located outside the boiler proper. The whole process of steam evaporation is effected outside the furnace, partly by "Flashing"

on a small reduction in pressure, and partly by evaporation produced by the flashed steam after it is superheated. A continuous blow-down from the external evaporator is maintained. It is claimed that this permits the use of heavily contaminated or even salt water feed. The addition of the evaporator adds weight to the unit which is undesirable, but also it increases somewhat the amount of feed in the system which is advantageous from the point of view of control.

The development of this boiler was at first rather slow. Only in 1936 after the announcement of the German Four Year Plan did the orders become numerous. As the Benson is principally a high-pressure boiler, and as at first high pressure was considered to be economical for large outputs only, most of the boilers ordered were large ones having outputs of 200,000 pounds per hour or more. Over 250 of these boilers have been and are being built, of which nearly one hundred were constructed for the German Navy.

B. Ramzin

This boiler was developed about 1930 by a Russian engineer and scientist named Professor L. K. Ramzin as a result of some research work in which he had been engaged. It is of the once through type and bears a resemblance to the Benson boiler.

Figure 14 shows a flow diagram of this boiler. As the diagram shows, the feed water is first passed through vertical tubes which also act as a suspension of the coiled surface in the first pass of the boiler. The water is heated during this passage and the warm water is then carried to the economizer. From the economizer the water is discharged into the

radiant evaporator section integral with the first radiant superheater section, both sections being in counterflow relative to the flue gas. The second and third superheater sections are arranged in parallel flow. The flue gas, after passing through the economizer, passes through the interstage reheater and then the air preheater.

Figure 2 shows another variation of this boiler. In this case, the feed water after passing through the economizer flows to a header at the lower end of the boiler furnace where it proceeds through coiled tubes in parallel flow to the roof of the furnace. Larger diameter tubes are used in the top than in the bottom of the furnace. The tubes are arranged to form a slag screen ahead of the superheater.

Critical control must be used, since water storage is small and the danger of burning a tube is ever present. It would seem that salts normally present in feed water would deposit in the upper portion of the furnace since this is the transition zone in Figure 2 or in the lower V-tubes or convection zone in Figure 14. Apparently some arrangement for eliminating salts similar to that used in the Benson boiler is utilized. The drop in pressure from economizer to superheater outlet is reported to be several hundred pounds.

Since this boiler is a Russian development, accurate information on it is scarce. It is reported that as far back as 1937, twenty-six boilers of this type were on order. A large unit for 330,000-440,000 pounds of steam per hour at 2840 psi and 896° - 932° F. was in use by 1938 and up to the time of the outbreak of World War II had given reasonably good service. This boiler was fired with pulverized coal.

An interesting side light on these boilers is the fact that Professor Ramzin was primarily interested in slag elimination and consequent furnace volume reduction, and his boiler development was purely a side line. In a lecture in 1938 Professor Ramzin described a boiler with two combustion chambers. In the first the fuel was burned almost completely, the ash melted and precipitated out into the slag bath on the chamber floor before the hot gases pass through screening tubes to the second combustion chamber. The heat release rate was very high in the first chamber and for both together amounted to about three times what was considered normal at that time with a reduction in overall boiler size of nearly fifty percent over "once through" boilers of similar capacity with dry ash removal. Whether this idea ever progressed out of the experimental stage is not known, but the idea appears feasible and may be in use with the present day Ramzin boiler.

C. Sulzer

The Sulzer Monotube Steam Generator was developed about 1930. It operates on the "once-through" principle and is a typical coil type boiler. It has all the advantages and disadvantages inherent in this boiler type. It resembles in principle the Benson and Ramzin designs, and except for the fact that there is no recirculation, it is similar to the Steamotive type. For high pressure and small steam outputs the Sulzer unit consists of a single coiled tube of considerable length. The required length of tubing is obtained by connecting the requisite number of tube lengths by means of welded joints of the design shown in Figure 15. In this design the tube ends are expanded and

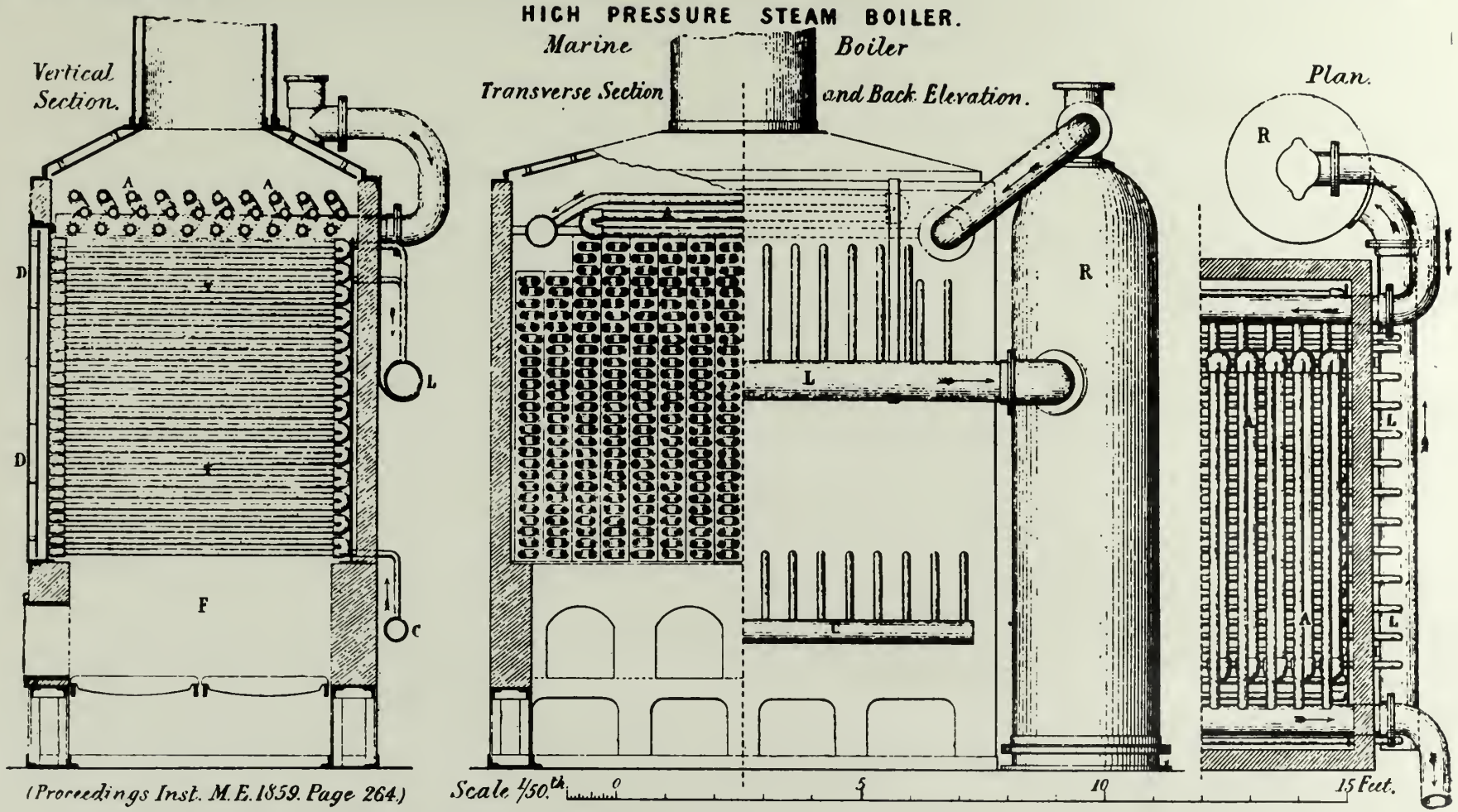
a nipple having a cone at each end, inserted. This prevents weld metal getting inside the tube and gives correct alignment of the tubes. The nipple tends to act as an orifice, but the pressure drop is small and disturbance helps to maintain turbulent flow which is beneficial to heat transmission.

In large units, several tube circuits are necessary in order to obtain the best operating conditions. This is also true in the case of low operating pressure in order to avoid excessive pressure drop in the boiler. A Sulzer unit operating at 500 psi and 44,000 pounds per hour steam output could be made up of four parallel circuits, each 2,500 feet long, made of 1.18 inch I.D. tubing and with a total heating surface of 5,380 square feet in the zone where evaporation is carried out. The superheater could be made of 2.05 inch I.D. tubing. The final velocity of the steam would be about 173 feet per second. The same requirements could be met with three tubes in parallel, each 3,300 feet long and with the same internal diameter. The pressure drops in these two designs would be 128 psi and 284 psi respectively. If operated along with a turbine the power loss for the 128 psi drop would be 0.18 per cent of the boiler output while for the 284 psi drop the loss would be o.4 per cent.

With a high pressure boiler of say 1500 psi the feed pressure is about 2000 psi and the power loss in the pump amounts to 2.7 per cent of boiler output. It is essential then to keep the pressure loss low especially as output pressure is increased. The real problem with this as well as other "once through" boilers is the need for quick-acting reliable controls which allows the unit to follow load

fluctuations without undue changes in boiler outlet pressure and temperature. Control cannot be based on the inlet feed water since the time required to traverse the boiler circuit may be several times the permissible delay time. Water injection at the zone where the steam becomes dry and saturated is resorted to to achieve control. The velocity through the superheater is much higher than that through the preceding evaporator section of the boiler so that the time of passage of this injected water to the boiler outlet is only a fraction of the time required to traverse the whole coil. This secondary feed, therefore, has almost an immediate influence on the temperature of the outlet steam.

The salt deposits in this boiler are eliminated by a salt intercepter as described in Chapter I. The size of this vessel will vary from 17 to 42 cubic feet. Figure 16 shows a typical Sulzer Monotube Salt Interceptor.



Martin Benson's forced circulation boiler with separator drum, circulating pump, and controlled water distribution. Designed in 1856.

FIGURE 1

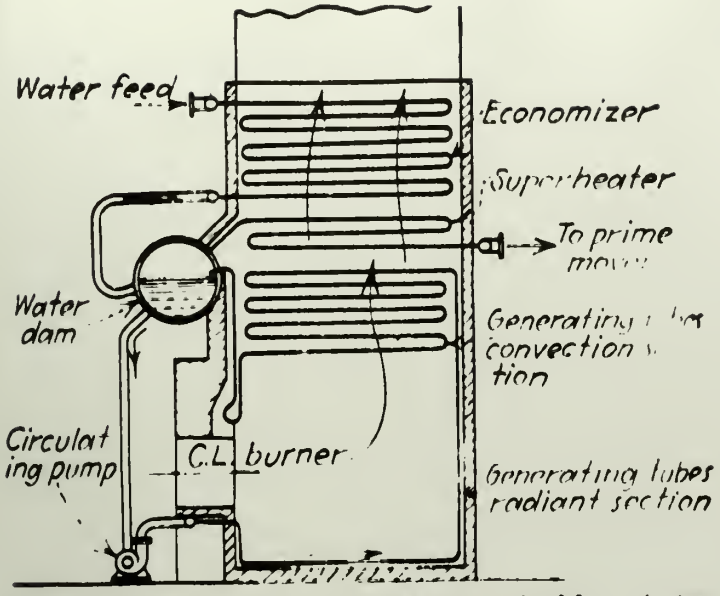


Fig. 3 Schematic arrangement, LaMont boiler

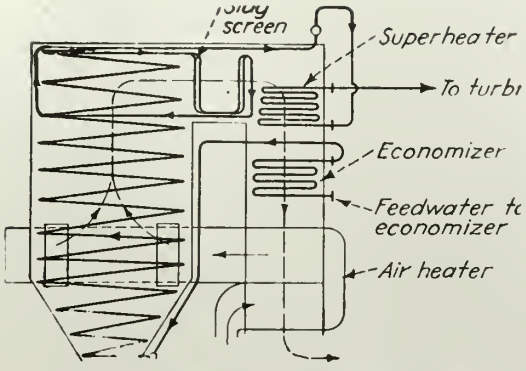


Fig. 2 Ramsin once-through boiler.

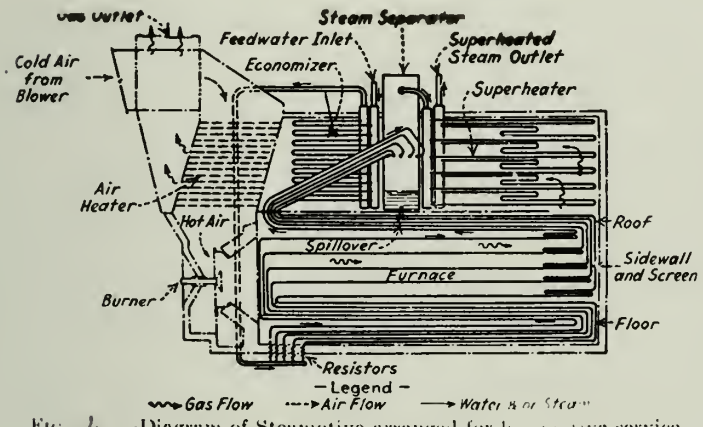


Fig. 4 Diagram of Steamotive arranged for locomotive service.

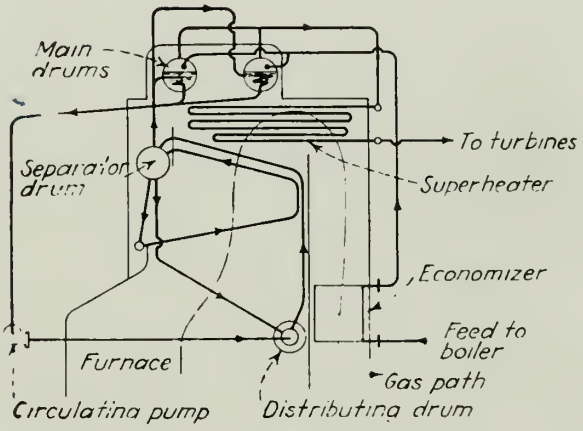
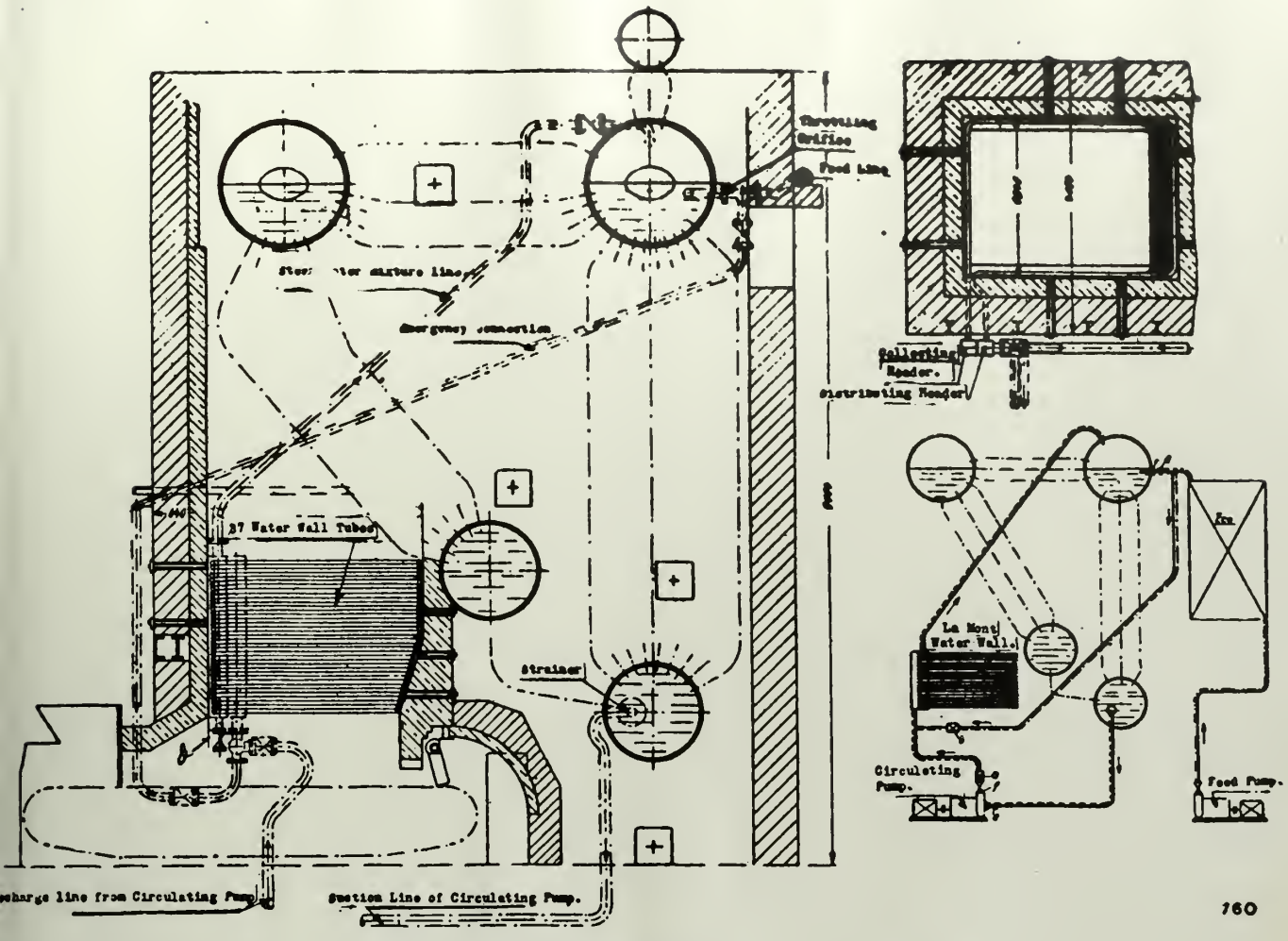


Fig. 5 Typical Schmidt-Hartman boiler.

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a, b = Check valves.
c, e, f = Shut-off valves.
d = Throttling orifice.

Fig. 6 —La Mont Water Wall fitted to a Bent Tube Water Tube Boiler.

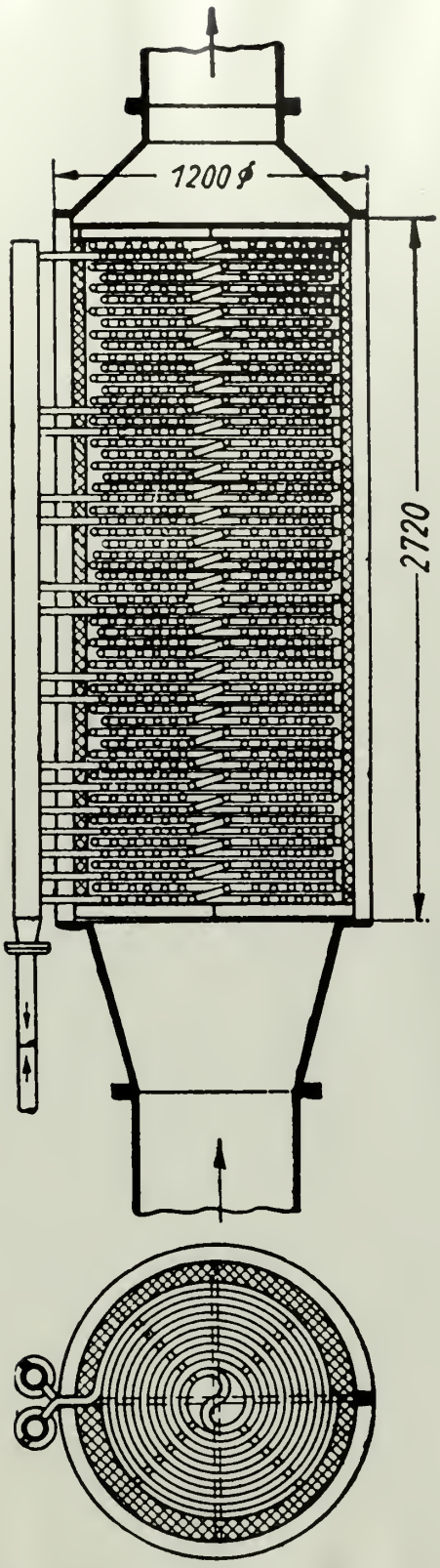


Fig. 7 —La Mont Diesel Exhaust Boiler.

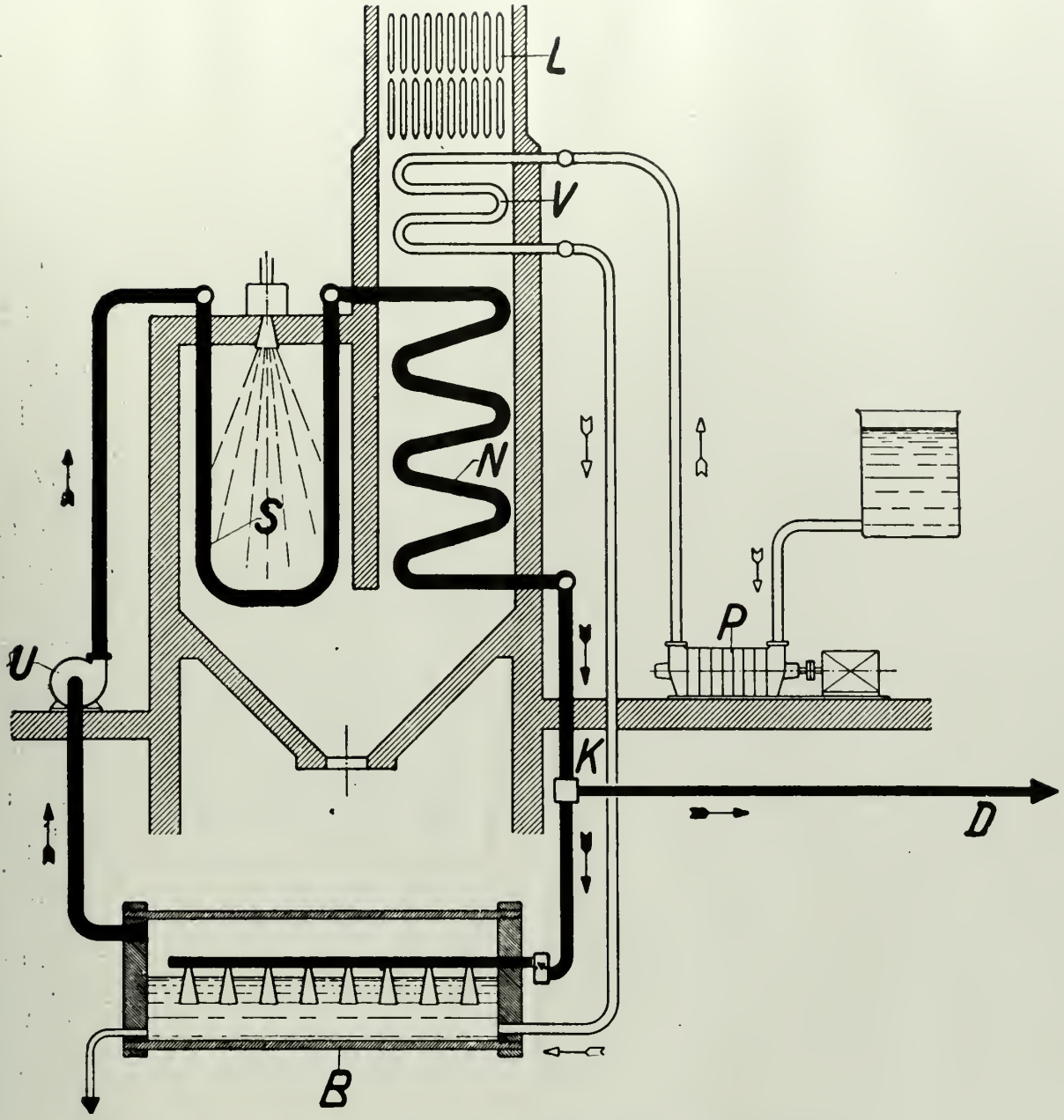
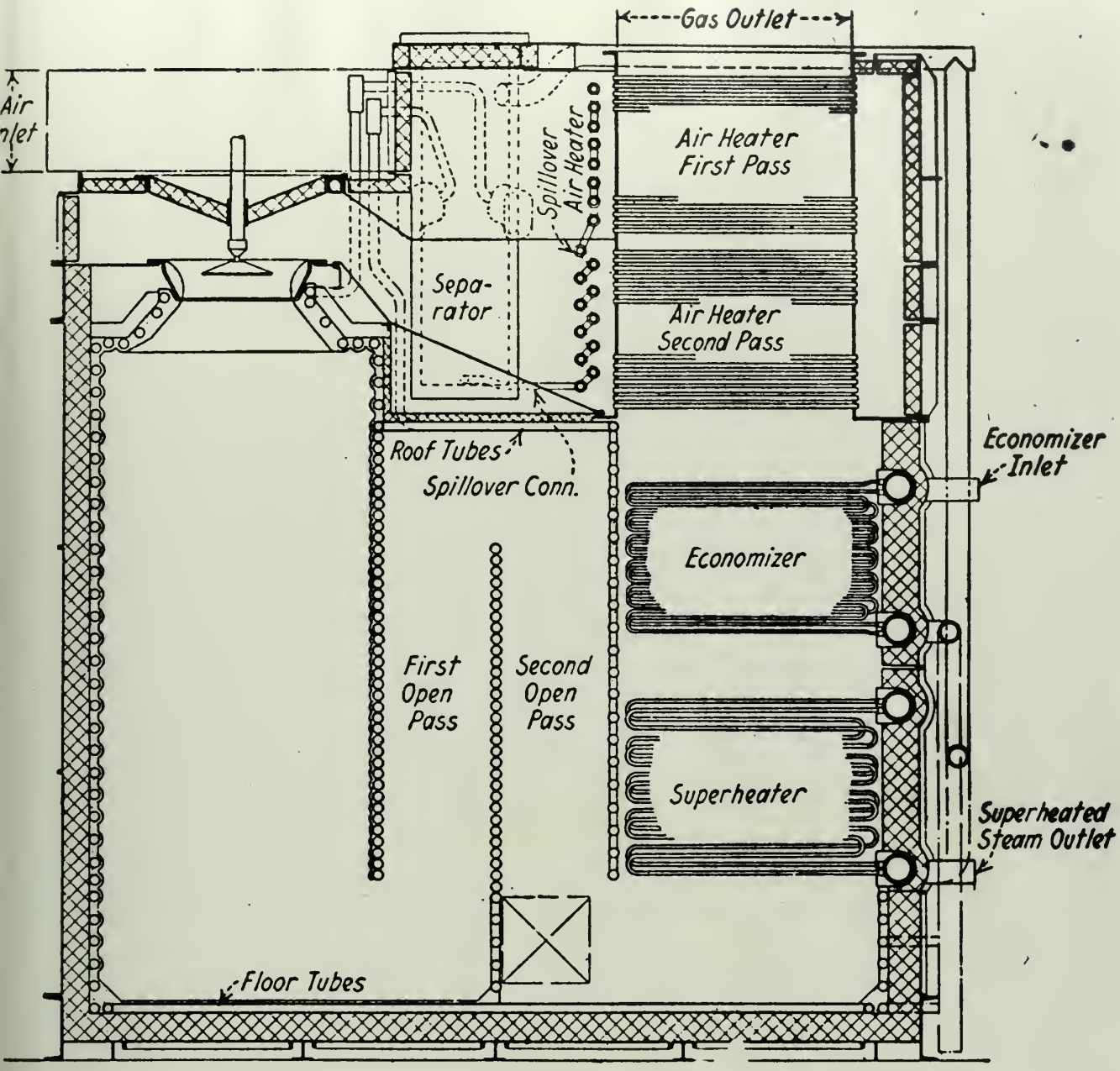


Fig. 8 —Principle of Loeffler Boiler.



Side Elevation – Section A-A

Fig. 9 —Open Pass Steamotive Unit.

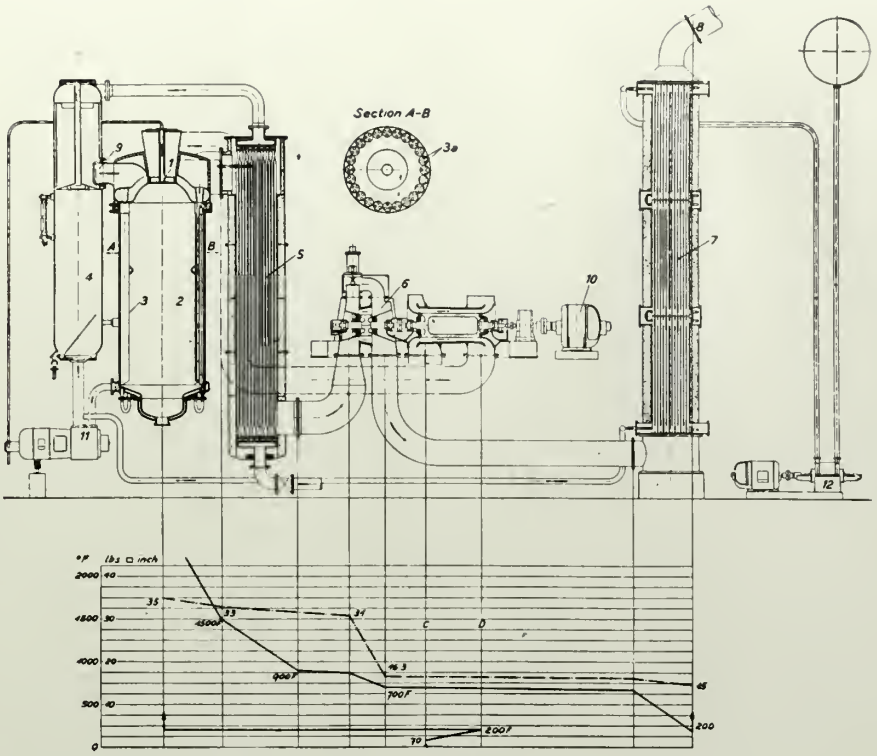


Fig. 10—Schematic View of Velox Unit.

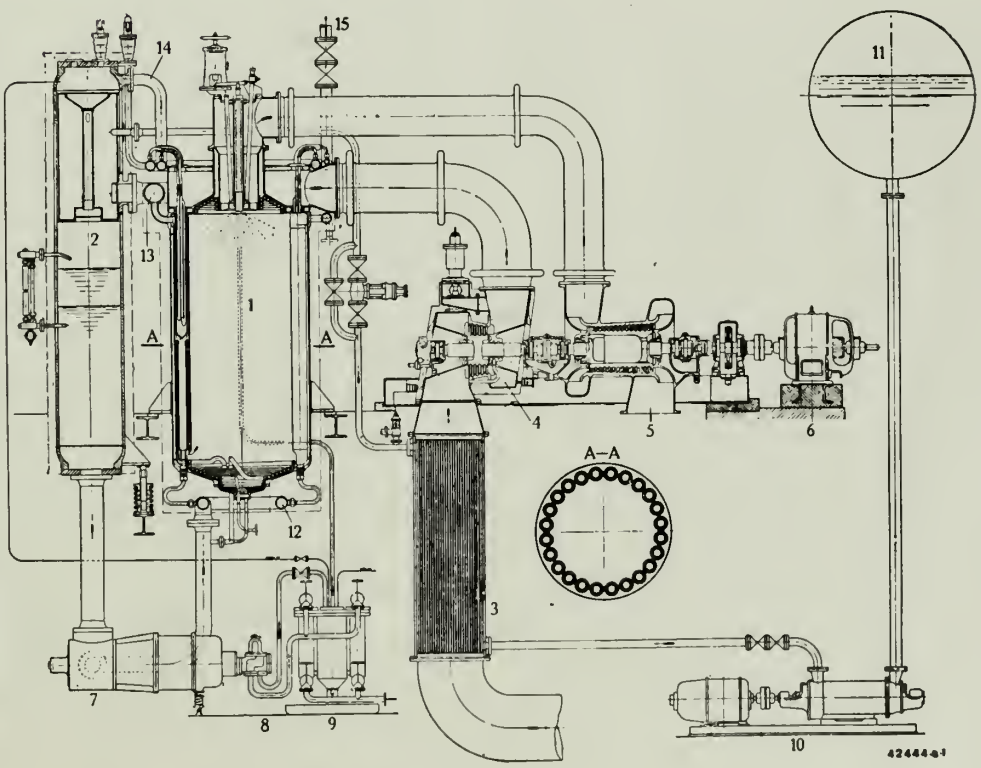


Fig. 11—Oil-fired Velox Steam Generator with Integral Superheater and Electrically-driven Auxiliaries.

1. Furnace with evaporator elements and integral superheater.
2. Steam separator drum.
3. Economiser (flue type).
4. Exhaust turbine } charging unit.
5. Air compressor }
6. Starting motor
7. Circulating pump.
8. Fuel oil and pressure oil pump.
9. Oil preheater and filter.
10. Feed pump.
11. Feed water tank.
12. Water distributing header.
13. Steam and water collecting header.
14. Sat. steam line to superheater.
15. Steam main.

a) Evaporator element.
b) " " outlet joint.
c) Alternative evaporator element outlet joint.
d) Evaporator element inlet joint.
→w = Water or water-steam mixture. →g = Flue gas.

FIGURE 12

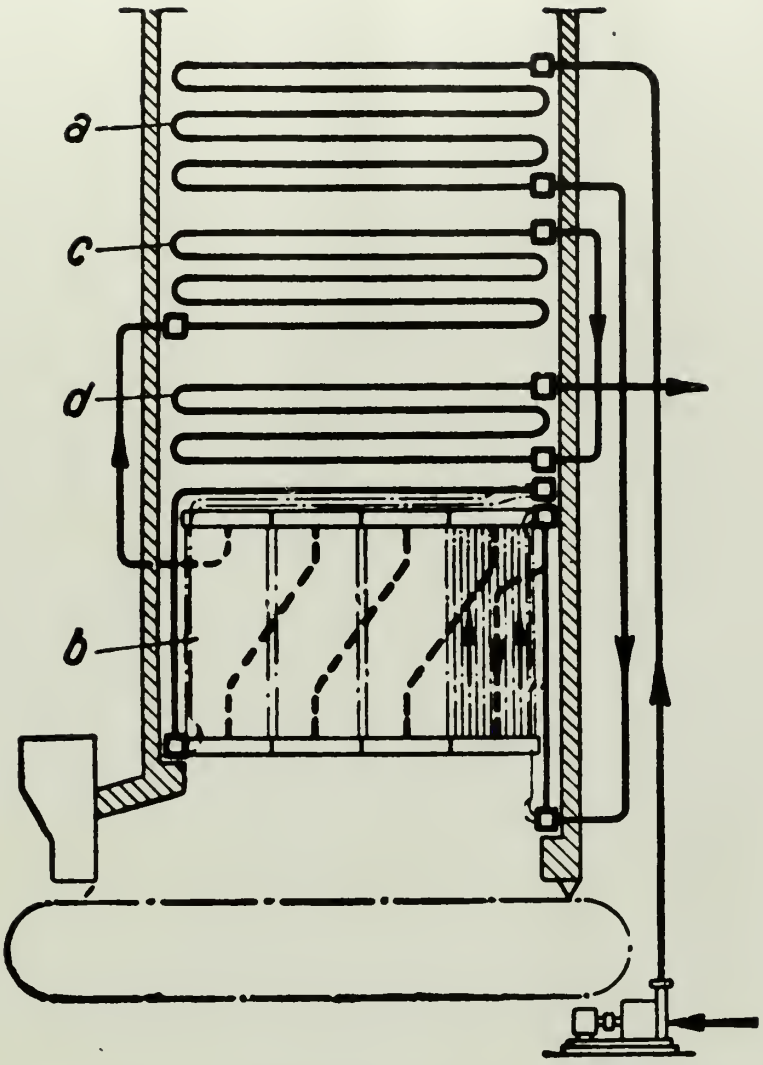


Fig. 13 —Flow Circuit of Small Benson Boiler.

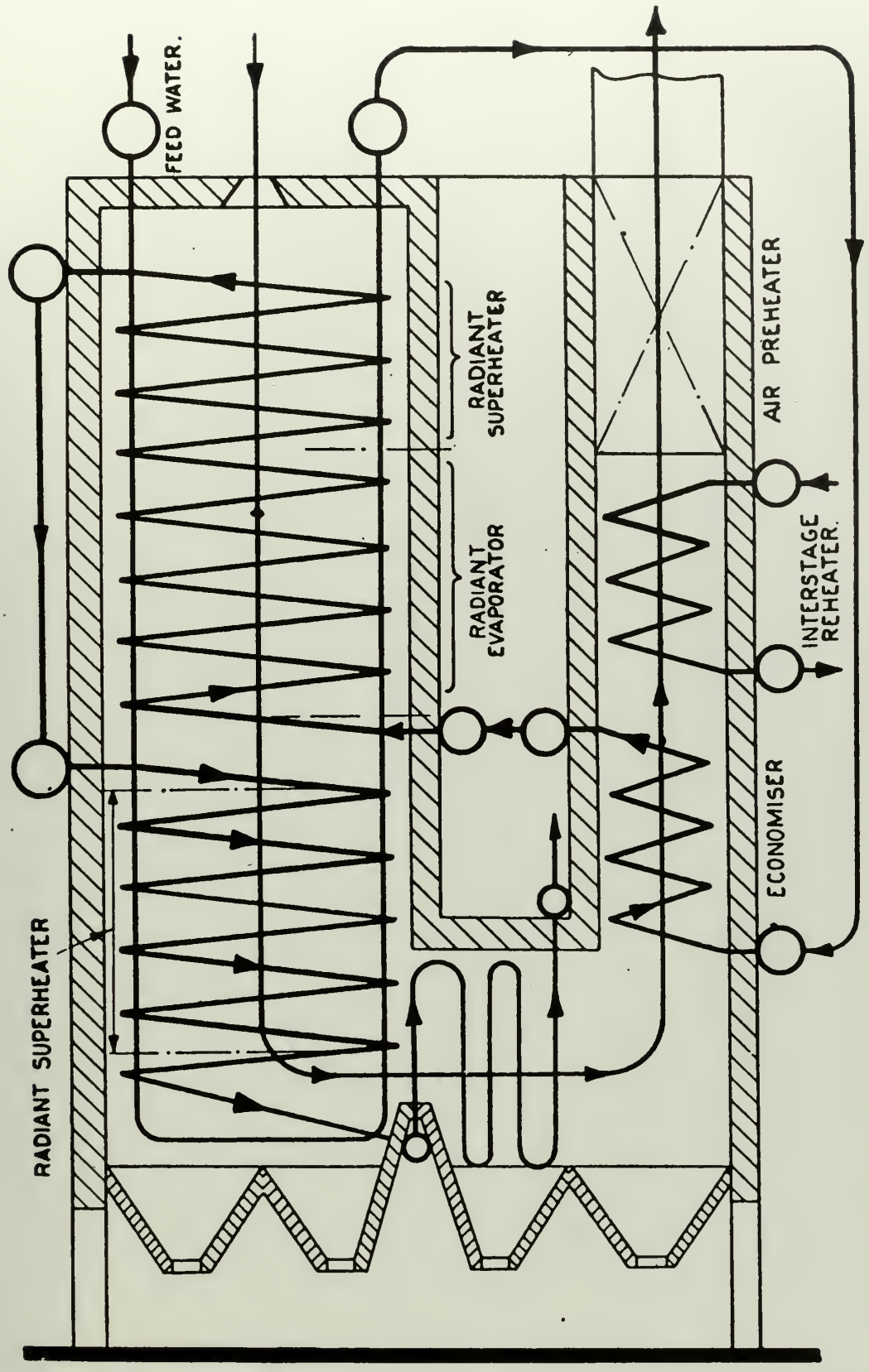


Fig. 14 —Flow Diagram of Ramzin Boiler.

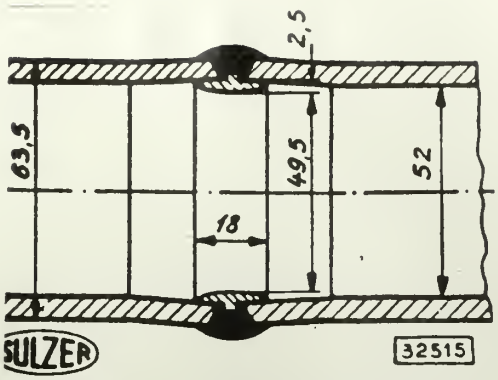


Fig. 15 —Sulzer Welded Tube Joint.



Fig. 16 —Sulzer Salt Interce
the Monotube Steam Gen

III. COMPARISON OF FORCED AND NATURAL CIRCULATION BOILERS

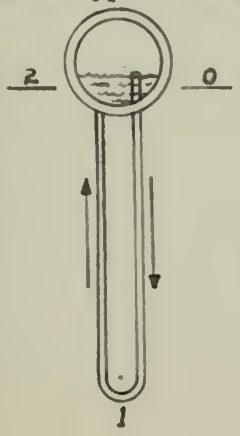
1. Circulation

A. Natural Circulation and its Problems

Natural circulation boilers are essentially similar in their modes of operation irrespective of whether they are of the bent-tube or straight-tube types. On the other hand, forced circulation boilers are of two rather distinctive main types. In this chapter, wherever possible, the natural and forced circulation boilers will be compared in general, but where necessary to show the proper picture, the comparison will be made with each kind of forced circulation individually.

Since the difference in circulation is the basic cause of all the other dissimilarities between these two kinds of boilers, it is logical to start with a description of the principles of natural circulation and the problems which result from it, followed by a discussion of forced flow as it pertains to the same conditions, along with its attendant problems. Other differences, of which there are a number, will then be compared.

In order to illustrate the principles of natural circulation the simplest type of evaporating circuit such as shown in the sketch will be used. This circuit has no restrictions in either the riser or downcomer leg and the tube is of uniform diameter. When heated on the riser leg this circuit will produce the maximum circulation, that is, the maximum weight of water and steam





through the circuit for a given head.

Any departure from this simple circuit which introduces resistances to flow such as headers, reduced tube areas, sharp bends, etc. will reduce the weight of water circulated. A reduction in height of downcomers, or heating of water in the downcomers with consequent steaming and reduction of density of the mixture entering the riser, will also cause a decrease in available head and flow.

The calculation of natural circulation is not an exact science since there are a number of factors involved which are difficult and sometimes impossible to determine. Several circulation theories have been proposed such as the "expansion" and "thermodynamic" theories. The theory which seems to give reasonably accurate results and which is the one most commonly used today considers natural circulation as a "pumping" process in which the static head available to overcome all resistances in the circuit depends upon the density of fluid in the risers and downcomers. The various factors involved in this method have been known for a number of years, but different designers applied varying weight factors to each of the terms. Within the last ten to fifteen years because of acceptance by many of the foremost boiler designers the method has become fairly standardized. Appendix "A" shows this theory as applied to a simple U-circuit and also shows how it stems from the General Energy Equation for flow.

The entrance loss is taken equal to one-half the velocity head at inlet. The velocity head in this case is based on the velocity and relative density at the inlet.

$$\frac{1}{2} \times \frac{U_s^2}{2g} \times \text{rel. density}$$

Some designers will double this to compensate for turbulence losses in headers or due to bifurcated or trifurcated tubes if such exist in the circuit.

The exit loss is taken as one velocity head. The velocity and density in this case is based on exit values.

$$\frac{U_E^2}{2g} \times \text{rel. density}$$

The friction loss comes from the Fanning equation and is based on midpoint velocity and density.

$$\frac{4fZU_m^2}{D \times 2g} \times \text{rel. density}$$

f = dimensionless factor = 0.006

U_m = midpoint velocity, feet per second

Z = tube height, feet

D = inside diameter of tube in feet

g = 32.2 feet per second per second

There exists some disagreement as to the proper value of f. At present 0.006 is appearing more frequently in the literature and is being used by the boiler manufacturers, even though 0.005 is the value most commonly seen. At best it is merely a conservative guess as to the relative roughness in boiler tubes and helps to account for some of the indeterminate factors in circulation. In this paper 0.006 will be used. Appendix "B" gives several examples of how this compares with actual values.

$$\frac{\quad}{\quad} = \quad \text{[illegible]}$$

[illegible] [illegible] [illegible] [

The acceleration loss is a function of the difference in velocity of the water at entrance and the velocity of wet steam at exit of the tube.

$$\frac{U_s}{g} (U_E - U_s)$$

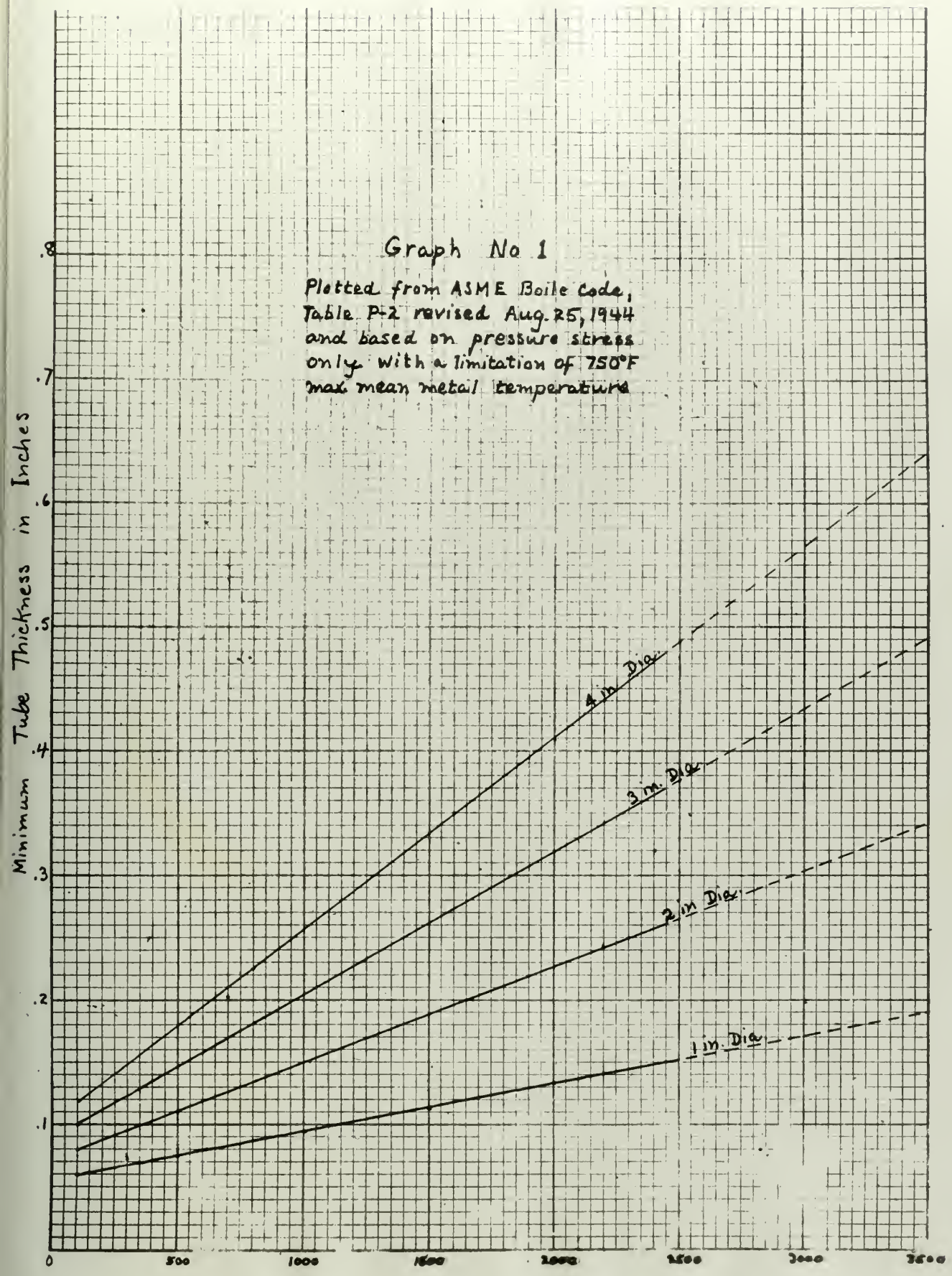
The criterion of adequate circulation is the complete protection of the tubes. Just what is necessary to obtain such complete protection is not positively known. It is generally believed that a mixture of 75 to 80% steam by volume will give ample protection to the top end of tubes at any pressure providing the velocity is such as to give turbulent flow. The tube angle and inside diameter also have a bearing on tube protection. Boiler designers generally assume that the tube is uniformly heated over its entire length and that there is no slip between the steam bubbles and the water. In the average boiler the rate of heat absorption is greater in the bottom portion of the furnace. Therefore, the assumption of uniform heat tends to give too high a value for the specific volume of the upper portion of the heated tubes. On the other hand, some slip velocity exists between the steam bubbles and the water. Moreover slag accumulations will be heavier in the zone of highest temperature which is the lower section of the furnace. The neglect of slip and the effects of slag tend to balance the inaccuracy of assuming uniform heat absorption. As a result uniform heat absorption is a realistic assumption in the average boiler.

The simple U-tube which is simpler than a boiler circuit will now be used to illustrate the limitations and problems inherent in natural

circulation. Calculations will be made for 500, 1000, 1500, 2000, 2500, 3000, and 3100 psi. These pressures are for saturated temperature conditions. The 500 psi calculation will be shown in detail, both to illustrate the method and to clarify certain steps along the way. For the other pressures the data will be given in tabulated form with accompanying graphs. Three items will be held constant

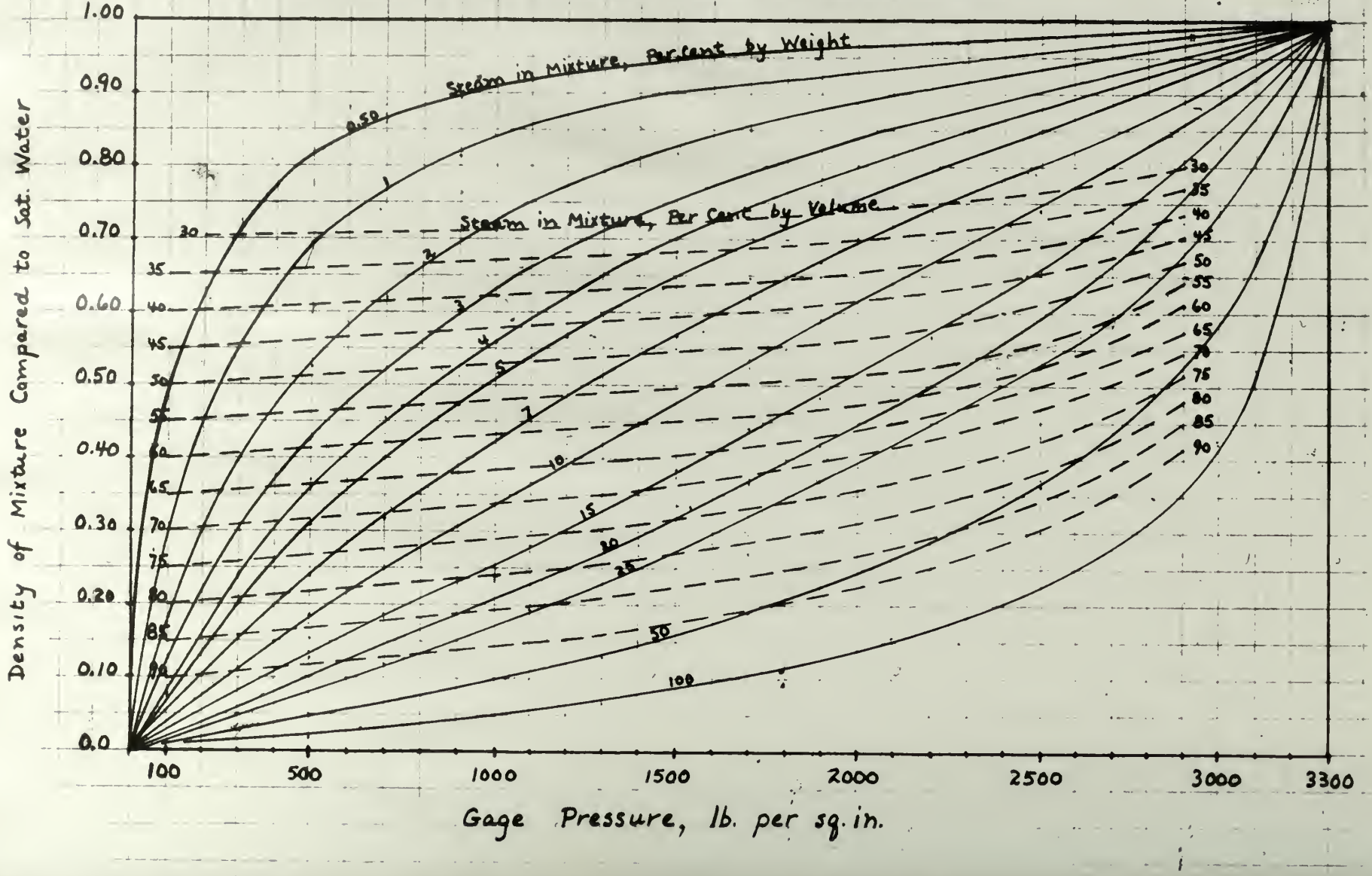
Tube height	=	60 feet
Inside tube diameter	=	1 inch
Constant heat absorption rate	=	70000 BTUs per sq ft. per hour.

It is assumed that there is no steaming in the downcomer, so that a solid flow of water enters the riser.


Graph No 1
Plotted from ASME Boile Code, Table P-2 revised Aug. 25, 1944 and based on pressure stress only with a limitation of 750°F max mean metal temperature
Minimum Tube Thickness in Inches
.8
.7
.6
.5
.4
.3
.2
.1
0
500
1000
1500
2000
2500
3000
3500
4 in Dia
3 in Dia
2 in Dia
1 in Dia

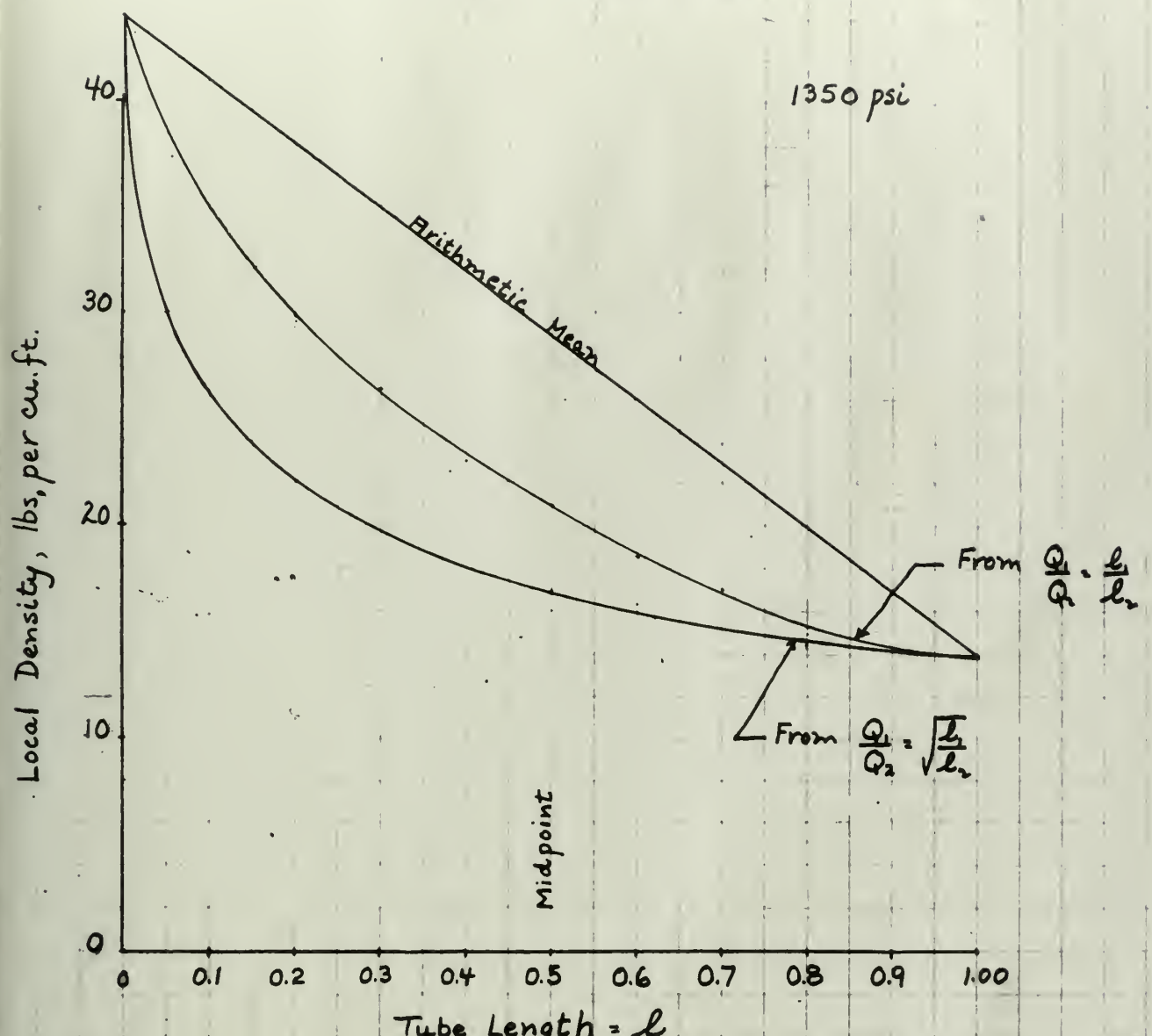
Graph No. 2

Note: This graph not accurate below 100 psi nor above 2900 psi



Graph No. 3

Changes in Density



Case I 500 psi and 467°F saturation temperature

From Graph 1 for the above conditions of pressure and temperature the tube thickness for one inch I.D. tube is .075 inch.

$$\therefore \text{ Outside Diameter} = 1.15 \text{ inches}$$

Projected area of boiler tube exposed to furnace heat:-

$$\frac{1.15}{12} \times 60 = 5.75 \text{ sq. ft.}$$

Total heat input to tube :-

$$5.75 \times 70{,}000 = 402{,}500 \text{ BTUs per hour}$$

From the Steam Tables the enthalpy of evaporation is equal to 755 BTUs per pound. The evaporation or mass rate of steam is:-

$$\dot{m} \text{ (steam)} = \frac{402{,}500}{755} = 533 \text{ pounds per hour}$$

Assume that there is 75% steam by volume leaving the top of the tube. From Graph 2 this gives a top dryness (percent by weight of steam at top of tube) of 0.06

$$\therefore \dot{m} \text{ (water)} = \frac{533}{0.06} = 8880 \text{ pounds per hour}$$

The mean density is then calculated. This is not the arithmetic mean, but rather is based on the condition of uniform heat absorption:

$$\frac{Q_1}{Q_2} = \frac{l_1}{l_2}$$

$$\frac{\sigma_1}{\sigma_2} = \frac{\lambda_1}{\lambda_2}$$

Q_1 = Heat absorbed by lowest portion l_1 of furnace wall tube

Q_2 = Heat absorbed by greater portion l_2 of furnace wall tube

l_1 and l_2 - see Q_1 and Q_2 above

Graph 3 shows a plot of the changes in density along the tube for saturated water at the bottom and a top dryness of 0.18, corresponding to a top density of 13.3 pounds per cubic foot which means a top density relative to water of 0.305. Curve A is a straight line giving an arithmetic mean density at midpoint. Curve B is based on uniform heat absorption while Curve C is based on:-

$$\frac{Q_1}{Q_2} = \frac{\sqrt{l_1}}{\sqrt{l_2}}$$

This curve is an example of a completely clean boiler which shows how density tends to vary if the heat absorption is non-uniform and is greater in the lower region of the tubes. Most designers use densities as indicated by Curve B, but there are a few who consider this too high and apply a so-called "experience factor" to arrive at densities between Curves B and C.

The true mean density of the mixture of steam and water in Curve B is given by the following empirical equation:-

$$d_m = \frac{W}{V^I} \ln \left(1 \neq \frac{1}{W}\right)$$

$$W = \frac{1}{X} \times \frac{V^1}{V'' - V^1}$$

d_m = True mean density, pounds per cubic feet

ln = Natural logarithm

X = Top dryness, percent

V^1 = Spec. volume of water at sat. temp. (Steam Tables)

V'' = Spec. volume of steam at sat. temp. (Steam Tables)

The true mean density of the steam - water mixture in this case is:-

$$W = \frac{1}{0.06} \times \frac{0.0197}{0.9278 - 0.0197}$$

$$W = \frac{0.0197}{0.0545}$$

$$W = 0.361$$

$$d_m = \frac{0.361}{0.0197} \ln \left(1 \neq \frac{1}{0.361} \right)$$

$$d_m = 18.33 \ln 3.77$$

$$d_m = 24.3 \text{ pounds per cubic foot}$$

From the Steam Tables $V^1 = 0.0197$ therefore the density in the downcomer and at entrance to the riser is:-

$$d_s = \frac{1}{V^1} = 50.8 \text{ pounds per cubic foot}$$

The available head from Appendix "A" is:-

$$\left(\frac{d_D}{d_s} - \frac{d_m}{d_s} \right) Z$$

$d_D = d_S$ since no steam in downcomer.

Therefore $(\frac{50.8}{50.8} - \frac{24.3}{50.8}) \times 60 = 31.3$ feet

If the assumption of top dryness = 0.06 is correct the sum of the losses in the circuit will just balance this static head. There are five losses in this simple circuit; entrance loss, friction loss in the downcomer, friction loss in the riser, acceleration loss in the riser, and exit loss. In order to calculate these losses it is first necessary to determine the velocity of the water U_s, the exit velocity U_E, and the midpoint velocity in the riser U_m. The first two of these are calculated from the Continuity equation in which the $\dot{m}$ and A are constant. The variation in velocity then is entirely dependent on change in density.

$\dot{m} = UAd$ (Continuity Equation)

$\dot{m}$ = mass rate, pounds per hour

U = velocity, feet per second

A = cross sectional tube area, square feet

d = density, pounds per cubic foot.

Therefore $U = \frac{\dot{m}}{Ad}$

$$U_s = \frac{8880}{3600 \times 50.8 \times .00545} = 8.92 \text{ ft. per sec.}$$

To get the velocity at riser exit, the density of the wet steam must first be determined. Using values of specific volume from the Steam Tables and the assumed top dryness of 0.06:-

$$V_E = V' + x\,(V'' - V')$$

$$V_E = 0.0197 + 0.06\,(0.9081)$$

$$V_E = 0.0742 \text{ cubic feet}$$

Therefore $d_E = \frac{1}{V_E} = 13.47$ pounds per cubic foot

Therefore $U_E = \frac{8880}{3600 \times 13.47 \times .00545} = 33.6$ ft. per second

Based on uniform heat absorption, an equal amount of steam will be generated in each foot of tube length. Therefore, the specific volume of the mixture and its velocity increase uniformly as a straight line function of tube length. The velocity head increases as the square of the velocity, but the relative density decreases in inverse proportion to the specific volume or velocity. Therefore, the product of velocity head times relative density will vary in direct proportion to the velocity or specific volume. Since friction loss is a function of this product, it is evaluated on the basis of an arithmetical average between the inlet and outlet velocities. This is the midpoint velocity and along with the midpoint relative density will be used to evaluate the friction loss in the riser.

$$U_m = \frac{U_s + U_E}{2} = \frac{8.92 + 33.6}{2}$$

$$U_m = 21.28 \text{ feet per second}$$

$$d_m = \frac{50.8 + 13.47}{2} = 32.13 \text{ pounds per cubic foot}$$

Using the above velocities the losses are: -

$$\frac{}{} = \frac{}{} = \frac{}{}$$

$$\frac{}{} = \frac{}{} = \frac{}{}$$

$$\frac{}{} = \frac{}{} = \frac{}{}$$

1. Entrance loss

$$\frac{1}{2} \times \frac{U_s^2}{2g} \times \text{rel. den.} = \frac{1}{2} \times \frac{(8.92)^2}{64.35} \times 1 = 0.62 \text{ ft.}$$

2. Friction loss in downcomer

$$\frac{4fZU_s^2}{D \times 2g} \times \text{rel. den.} = \frac{4(.006)(60)(8.92)^2}{.0833 \times 64.35} \times 1 = 21.40 \text{ ft.}$$

3. Friction loss in riser

$$\frac{4fZU_m^2}{D \times 2g} \times \text{rel. den.} = \frac{4(.006)(60)(21.28)^2}{.0833 \times 64.35} \times \frac{32.13}{50.8} = 77.2 \text{ ft.}$$

4. Acceleration loss in riser

$$\frac{U_s}{g}(U_E - U_s) = \frac{8.92}{32.2}(33.6 - 8.92) = 6.85 \text{ ft}$$

5. Exit loss

$$\frac{U_E^2}{2g} \times \text{rel. den.} = \frac{(33.6)^2}{64.35} \times \frac{13.47}{50.8} = 4.66 \text{ ft}$$

The sum of the losses is:-

$$.62 + 21.40 + 77.2 + 6.85 + 4.66 = 110.73 \text{ ft}$$

Available Head - 31.30 feet

Losses - 110.73 feet

The losses greatly exceed the available head which indicates that the assumption of top dryness was too low. So assume several larger values and repeat the steps above. This will give:-

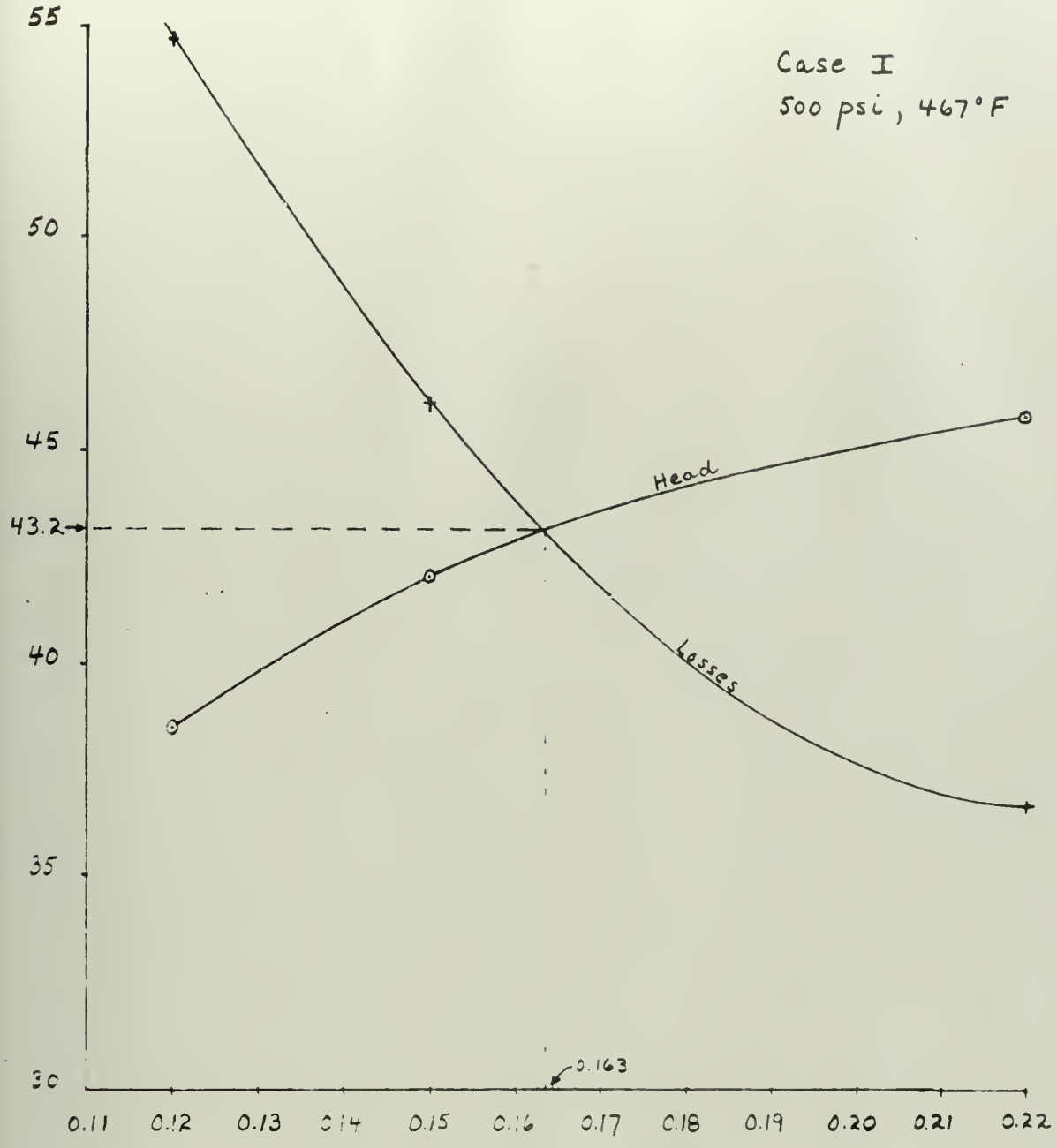
Top dryness	0.12	0.15	0.22
Mass rate of water, pounds per hour	4441	3553	2425
Density of water, pounds per cubic foot	50.8	50.8	50.8
Mean density of mixture, pounds per cubic foot	18.2	15.25	12.1
Density of exit mixture, lbs. per cubic foot	7.77	6.42	4.55
Midpoint density in riser, lbs. per cubic foot	29.28	28.61	27.67
Velocity of water, feet per second	4.45	3.565	2.43
Midpoint velocity in riser, ft. per second	16.78	15.91	14.82
Velocity at exit, feet per second	29.12	28.25	27.2
Available head, feet	38.5	42.0	45.8
Total losses, feet	54.7	46.1	36.7

These three points are plotted feet of head against top dryness as in Graph 4. If a curve is now drawn through the plotted heads, and another through the losses, these intersect at the equilibrium point which gives the top dryness and head necessary for stable operation. With this exit steam condition known, the other pertinent data for 500 psi may be computed. See summary at end of these calculations.

Low Common	0.22	0.11	1.00
Best Case of Public, Power and Light	4.16	3.41	3.61
Deputy or Water, Public Car Parts Fuel	0.9	0.4	0.4
Duty Counsel of Delivery, Ground Gas Motor Fuel	1.2	1.25	1.1
Deputy of Coal Product, Inc. Gas Motor Fuel	1.0	0.8	1.0
Regional Energy in Glass, Inc. Gas Motor Fuel	0.25	0.3	0.2
Delivery of Water, Fuel Gas Motor	1.1	1.0	1.1
Laboratory Machine in Beer, Inc. Gas Natural	1.4	0.9	1.2
Delivery of Gold, Coal Gas Natural	0.12	0.09	0.2
Available Beer, Fuel	0.5	0.0	0.1
Total Income, Coal	0.7	0.3	0.7

Three three points are plotted from a hand control the Income as in Graph 1. If a curve is now drawn through the Hobby Bank, and another through the Income, these tabernacle of the equilibrium point within close the top Income and hand necessary for ability operation. This rate will show condition Income, the other predicament data for hand pal may be computed. The accuracy at one of these calculations.

Graph No. 4


55
50
45
43.2
40
35
30
Case I
500 psi, 467°F
Head
Losses
0.163
0.11
0.12
0.13
0.14
0.15
0.16
0.17
0.18
0.19
0.20
0.21
0.22
Top Dryness

Case II 1000 psi and 544.6°F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I. D. tube is .095 inch.

$\therefore$ Outside diameter = 1.19 inches

Using method of Case I with three different assumptions for top dryness the following data is obtained:-

Projected area of tube	5.95 feet
Total heat input to tube	416,000 BTUs
Enthalpy of evaporation	649.4 BTUs per lb.
Mass rate of steam	641 lbs per hour

Top dryness	0.13	0.15	0.16
Mass rate of water, pounds per hour	4935	4280	4010
Density of water, lbs. per cubic foot	46.3	46.3	46.3
Mean density of mixture, lbs. per cubic foot	22.95	21.65	21.0
Density of exit mixture, lbs. per cubic foot	13.0	11.74	11.2
Midpoint density in riser, lbs per cu. ft.	29.65	29.02	28.75
Velocity of water, feet per second	5.41	4.73	4.43
Midpoint velocity in riser, ft. per second	12.38	11.66	11.36
Velocity at exit, feet per second	19.35	18.6	18.29
Available head, feet	30.25	32.0	32.8
Total losses, feet	38.56	32.5	30.15

Graph 5 is the equilibrium diagram for this data.

Then Design 1 is given preference over Base Economy Type 1 and Case 1.

Case 12 ... [illegible] and [illegible] Economy vs Conservative

Case 12 is the given condition for Base Economy Type 1 and Case 1.

Case 12 .005 Brass.

∴ Annular diameter = 1.327 inches

Using method of Case 1 with these different connections for Gap Bypass the following data is recorded:-

Projected area of tube	5.93 Sqms
Total heat input to tube	115, 800 BTUs
Rateability of evaporation	6674.4 BTUs per lb.
Mass rate of steam	640 lbs per hour

Fog degrees	0.13	0.35	0.36
Mass rate of water, pounds per hour	4775	4230	4010
Density of water, lbs. per cubic foot	45.3	44.4	43.3
Mean density of mixture, lbs. per cubic foot	22.85	21.65	22.3
Density of wall mixture, lbs. per cubic foot	32.0	31.74	32.2
Weighted density in piers, lbs per cu. ft.	27.69	27.32	21.27
Velocity of water, feet per second	7.23	6.70	6.43
Velocity in piers, ft. per second	12.34	11.44	10.36
Velocity at exit, feet per second	19.35	18.4	18.27
Available head, feet	30.23	30.1	30.4
Total losses, feet	26.51	26.1	30.14

Graph 5 is the equilibrium diagram for this case.

Case III 1500 psi and 596.2°F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I.D. tube is 0.113 inch.

$\therefore$ Outside diameter = 1.226 inches

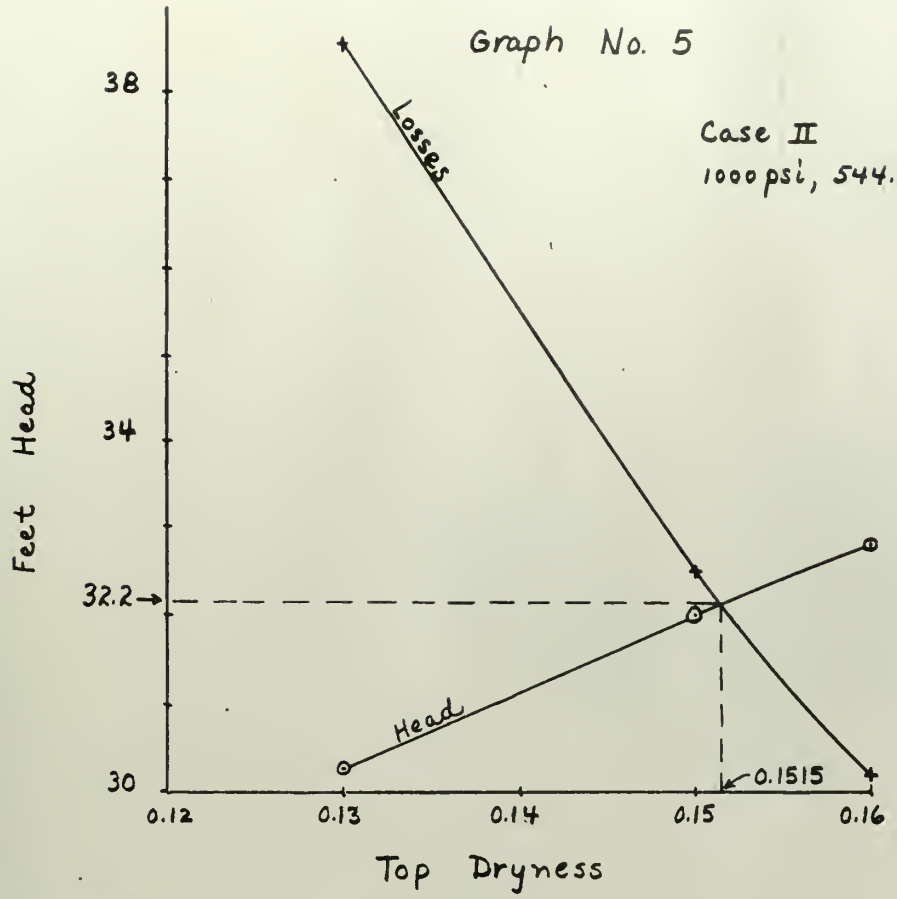
Using computations as in Case I with four different assumptions for top dryness the following data is obtained:-

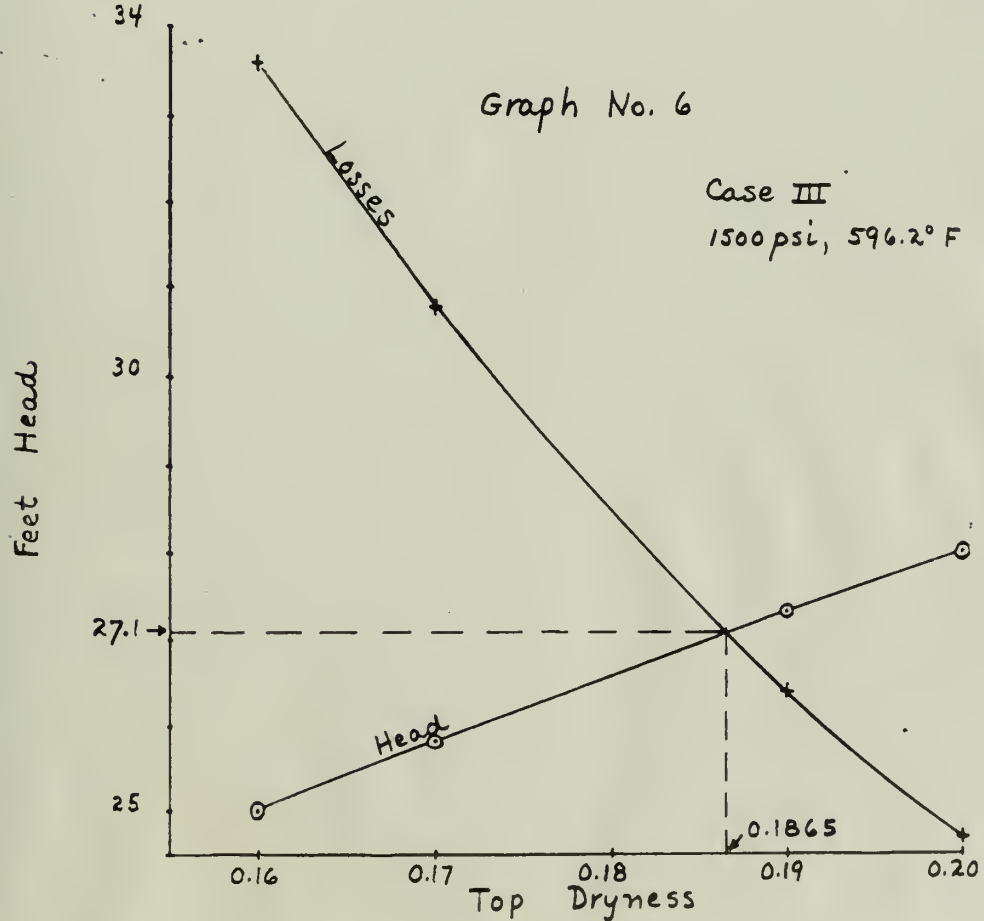
Projected area of tube	6.12 square feet
Total heat input to tube	429,000 BTUs
Enthalpy of evaporation	556.3 BTUs per lb.
Mass rate of steam	770 lbs. per hour

Top dryness	0.16	0.17	0.19	0.20
Mass rate of water, lbs. per hour	4805	4530	4050	3850
Density of water, lbs. per cu. ft.	42.55	42.55	42.55	42.55
Mean density of mixture, lbs. per cu. ft.	24.8	24.3	23.2	22.8
Density of exit mixture, lbs. per cu. ft.	15.62	15.05	13.96	13.5
Midpoint density in riser, lbs. per cu. ft.	29.1	28.8	28.25	28.0
Velocity of water, ft. per second	5.77	5.43	4.86	4.62
Midpoint velocity in riser, ft. per sec.	10.73	10.4	9.83	9.59
Velocity at exit, ft. per sec.	15.7	15.38	14.8	14.55
Available head, feet	25.0	25.8	27.3	28.0
Total losses, feet	33.6	30.8	26.4	24.7

Graph 6 is the equilibrium diagram for this data.

Graph 6 is the equilibrium diagram for this data.


Graph No. 5
Case II
1000 psi, 544.6°F
Losses
Head
38
34
32.2
30
Feet Head
0.12
0.13
0.14
0.15
0.16
0.1515
Top Dryness


Graph No. 6
Case III
1500 psi, 596.2° F
Losses
Head
34
30
27.1
25
Feet Head
0.16
0.17
0.18
0.19
0.20
0.1865
Top Dryness

Case IV 2000 psi and 635.8°F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I. D. tube is 0.133 inch.

$\therefore$ Outside diameter = 1.266 inches

Using computations as in Case I with three different assumptions for top dryness the following data is obtained:-

Projected area of tube	6.32 square feet
Total heat input to tube	443,000 BTUs
Enthalpy of evaporation	463.4 BTUs per lb.
Mass rate of steam	956 lbs. per hour

Top dryness	0.23	0.25	0.27
Mass rate of water, lbs. per hour	4160	3820	3540
Density of water, lbs. per cu. ft.	38.9	38.9	38.9
Mean density of mixture, lbs. per cu. ft.	24.1	23.9	22.7
Density of exit mixture, lbs. per cu. ft.	15.9	15.1	14.4
Midpoint density in riser, lbs. per cu. ft.	27.4	27.0	26.65
Velocity of water, ft. per second	5.45	5.01	4.64
Midpoint velocity in riser, ft. per sec.	9.43	8.96	8.59
Velocity at exit, ft. per second	13.4	12.92	12.55
Available head, feet	22.75	23.9	24.9
Total losses, feet	27.58	24.2	21.6

Graph 7 is the equilibrium diagram for this data.

Case IV. Flow in a 24" diameter concrete pipeline

For Area, for above conditions Col 12b, Table 6.5 = 0.785

I. D. Tube is 0.329 feet.

∴ Outside diameter = 1.314 inches

Using comparisons as in Case I with these different assumptions the top drymass line following data is obtained:-

Projected area of tube	5.38 square feet
Total mass input to tube	340,000 lb.
Loading of suspension	143.4 lbs per ft.
Mass rate of mass	324 lbs. per sec.

Top drymass	0.23	0.23	0.23
Mass rate of water, lbs. per hour	4130	3720	3600
Density of water, lbs. per cu. ft.	74.9	59.4	57.4
Mean density of mixture, lbs. per cu. ft.	91.3	83.4	82.7
Density of wet mixture, lbs. per cu. ft.	31.9	11.2	11.4
Maximum density in sieve, lbs. per cu. ft.	87.4	87.4	84.4
Velocity of water, ft. per second	1.43	5.01	4.8
Maximum velocity in sieve, ft. per sec.	1.41	4.80	4.57
Velocity at exit, ft. per second	11.4	12.8	12.45
Available head, feet	80.2	21.9	84.4
Total losses, feet	47.21	84.2	24.4

Case V is the specification figures for this case.

Case V 2500 psi and 668.1° F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I. D. tube is 0.1525 inch.

$\therefore$ Outside diameter = 1.305 inches

Using computations as in Case I with three different assumptions for top dryness the following data is obtained:-

Projected area of tube	6.525 sq. ft.
Total heat input to tube	456,500 BTUs
Enthalpy of evaporation	360.5 BTUs per lb.
Mass rate of steam	1269 lbs. per hour

Top dryness	0.36	0.38	0.40
Mass rate of water, lbs. per hour	3525	3338	3170
Density of water, lbs. per cubic foot	34.8	34.8	34.8
Mean density of mixture, lbs. per cu. ft.	22.45	22.05	21.72
Density of exit mixture, lbs. per cu. ft.	15.2	14.8	14.35
Midpoint density in riser, lbs. per cu. ft.	25.0	24.8	24.6
Velocity of water, ft. per sec.	5.16	4.95	4.65
Midpoint velocity in riser, ft. per sec.	8.5	8.3	7.97
Velocity at exit, ft. per sec.	11.84	11.65	11.29
Available head, feet	21.3	21.9	22.55
Total losses, feet	23.4	21.93	19.86

Graph 8 is the equilibrium diagram for this data.

Refer to Table 1 for more detailed information on the [illegible]

i. e. value in C.I.F. form.

$\therefore$ Average Elevation = 1,[illegible] [illegible]

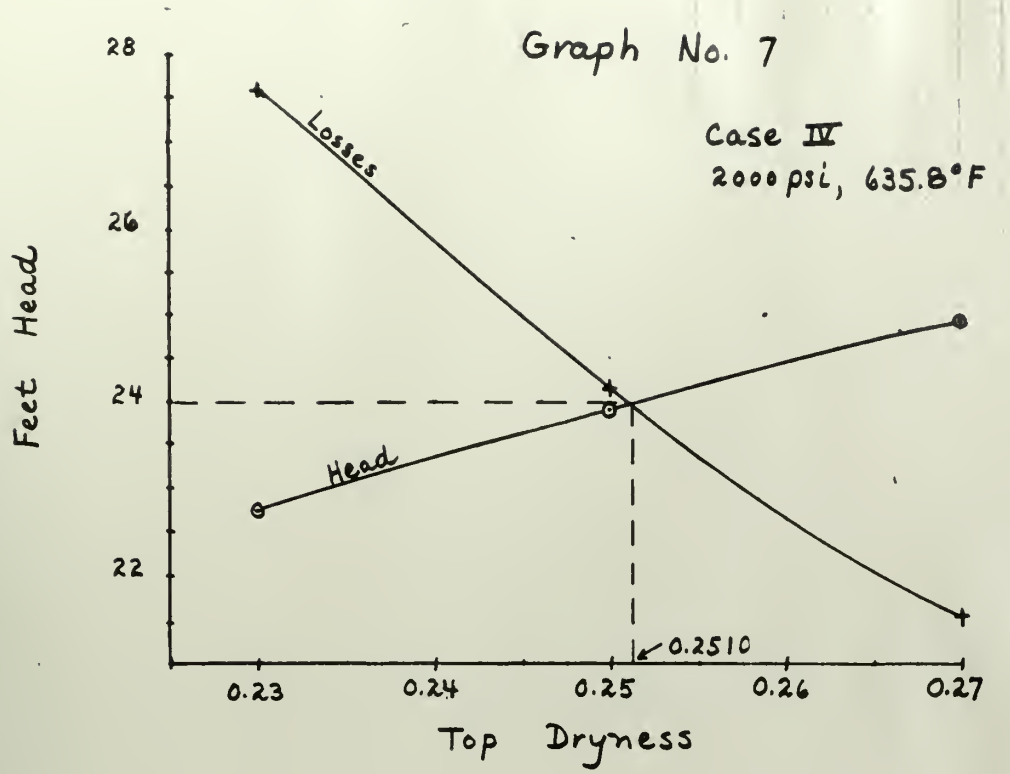
Using these definitions it is easy to write down directly the [illegible]

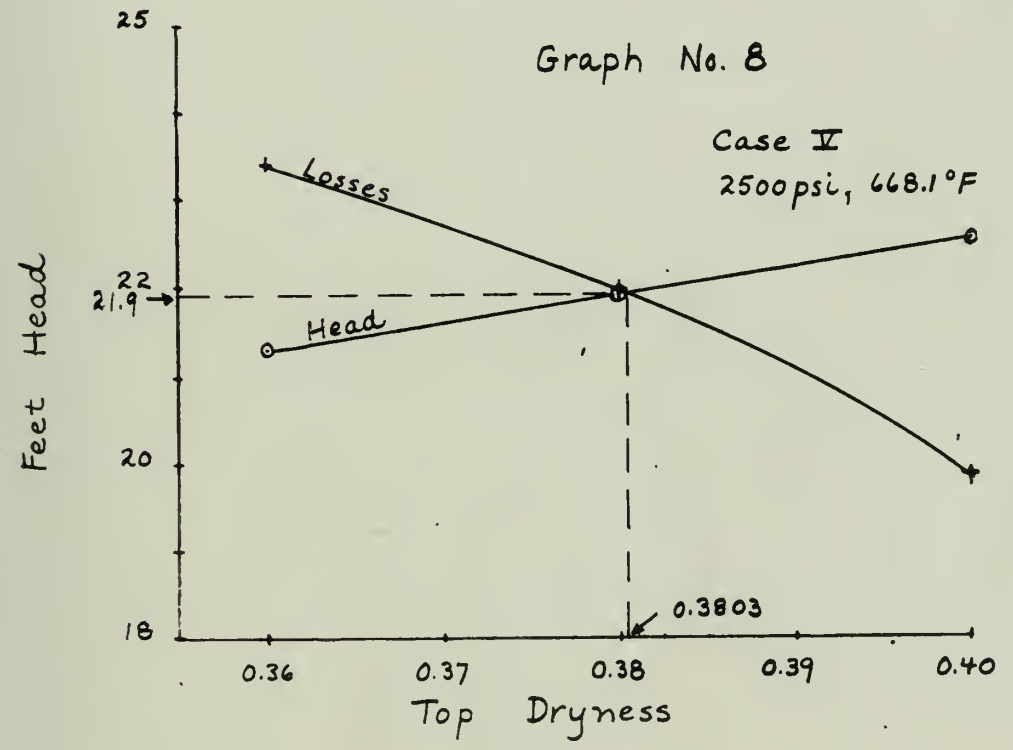
log arguments and additional data is obtained:-

Production area of [illegible]	6,[illegible] sq. ft.
Fuel [illegible] to [illegible]	[illegible] cubic [illegible]
Variation of [illegible]	23[illegible] miles per [illegible].
Mean [illegible] of [illegible]	[illegible] lbs. per hour.

Top dresses	6,[illegible]	[illegible]	[illegible]
Mean rate of water, lbs. per hour	[illegible]	[illegible]	[illegible]
Density of [illegible], 10[illegible] [illegible] [illegible]	[illegible]	[illegible]	[illegible]
Mean density of [illegible], lbs. per cu. ft.	[illegible]	[illegible]	[illegible]
Density of [illegible] [illegible], lbs. per cu. ft.	[illegible]	[illegible]	[illegible]
Original [illegible] to river, ton. per sq. ft.	[illegible]	[illegible]	[illegible]
Velocity of water, ft. per sec.	[illegible]	[illegible]	[illegible]
Original velocity in river, ft. per sec.	[illegible]	[illegible]	[illegible]
Velocity at exit, ft. per sec.	[illegible]	[illegible]	[illegible]
Available head, feet	[illegible]	[illegible]	[illegible]
Total losses, feet	[illegible]	[illegible]	[illegible]

Graph 2 is the equilibrium diagram for this case.


Graph No. 7
Case IV
2000 psi, 635.8°F
Losses
Head
28
26
24
22
Feet Head
0.2510
0.23
0.24
0.25
0.26
0.27
Top Dryness


Graph No. 8
Case V
2500 psi, 668.1°F
Losses
Head
25
22
21.9
20
18
Feet Head
0.3803
0.36
0.37
0.38
0.39
0.40
Top Dryness

Case VI 3000 psi and 695.4°F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I. D. tube is 0.17 inch.

$\therefore$ Outside diameter = 1.34 inches

Using computations as in Case I with three different assumptions for top dryness the following data is obtained:-

Projected area of tube	6.7 sq. ft.
Total heat input to tube	469,500 BTUs
Enthalpy of evaporation	217.8 BTUs per lb.
Mass rate of steam	2155 lbs. per hour

Top dryness	0.75	0.77	0.79
Mass rate of water, pounds per hour	2870	2800	2728
Density of water, lbs. per cu. ft.	28.9	28.9	28.9
Mean density of mixture, lbs. per cu. ft.	19.5	19.33	19.2
Density of exit mixture, lbs. per cu. ft.	13.7	13.51	13.3
Midpoint density in riser, lbs. per cu. ft.	21.3	21.2	21.1
Velocity of water, ft. per sec.	5.1	4.94	4.82
Midpoint velocity in riser, ft. per sec.	7.9	7.74	7.64
Velocity at exit, ft. per sec.	10.7	10.55	10.46
Available head, feet	19.44	19.85	20.1
Total losses, feet	21.15	20.27	19.5

Graph 9 is the equilibrium diagram for this data.

Prof. Smith's [illegible] Table 1 Solutions [illegible]

Q. 7. Show it [illegible].

$\therefore$ [illegible] = 1.34 [illegible]

Below [illegible] to Case I with [illegible]:-

[illegible] area of [illegible]	[illegible] sq. ft.
[illegible] [illegible] to [illegible]	[illegible]
[illegible] of [illegible]	[illegible] ft.
[illegible] [illegible]	[illegible]

[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]

Graph 7 in the [illegible] [illegible].

Case VII 3100 psi and 700.3°F saturation temperature

From Graph 1 for above conditions the tube thickness for a one inch I.D. tube is 0.17 inch.

$\therefore$ Outside diameter = 1.35 inches

Using computations as in Case I with three different assumptions for top dryness the following data is obtained:-

Projected area of tube	6.75 sq. ft.
Total heat input to tube	472,500 BTUs
Enthalpy of evaporation	168.1 BTUs per lb.
Mass rate of steam	2810 lbs. per hour

Top dryness	1.02	1.05	1.10
Mass rate of water, lbs. per hour	2753	2678	2550
Density of water, lbs. per cu. ft.	26.95	26.95	26.95
Mean density of mixture, lbs. per cu. ft.	18.42	18.32	18.04
Density of exit mixture, lbs. per cu. ft.	13.14	12.95	12.64
Midpoint density in riser, lbs. per cu. ft.	20.05	19.95	19.8
Velocity of water, ft. per sec.	5.21	5.07	4.83
Midpoint velocity in riser, ft. per sec.	7.96	7.81	7.56
Velocity at exit, ft. per sec.	10.7	10.55	10.3
Available head, feet	18.95	19.20	19.8
Total losses, feet	21.96	20.98	19.4

Graph 10 is the equilibrium diagram for this data.

Therefore I can draw conclusions by the following for a new [illegible]
from it [illegible].

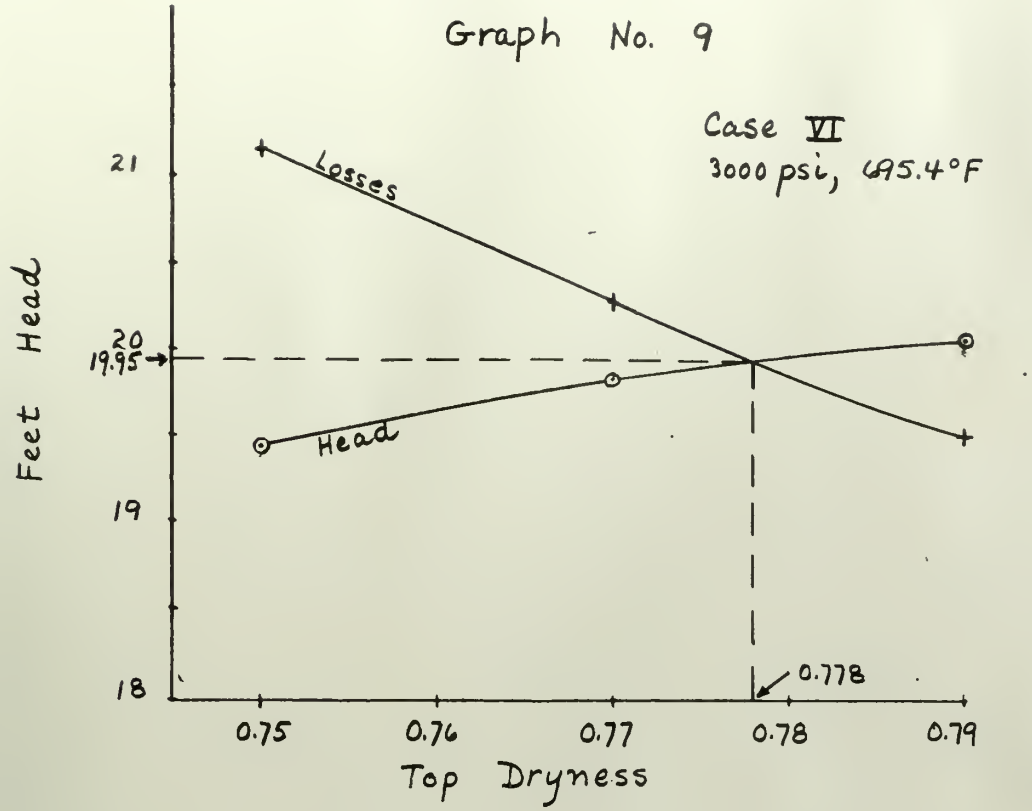
∴ [illegible] density = 1.25 grams

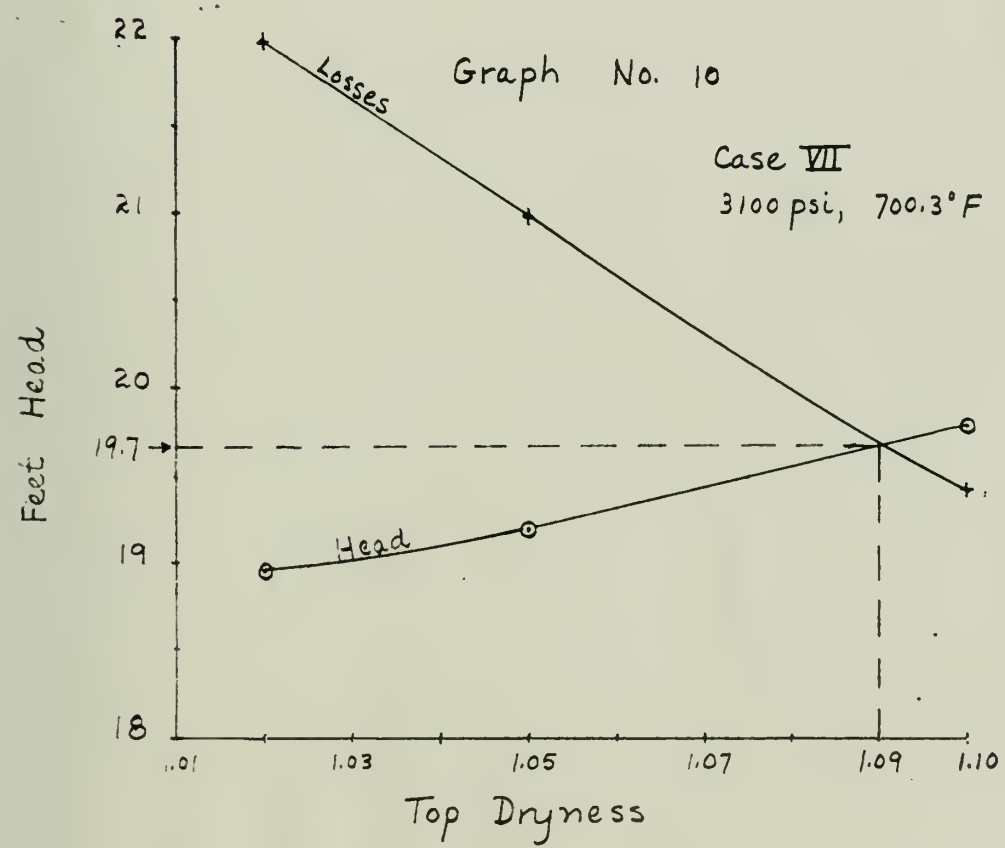
Many [illegible]ions as to [illegible] there different [illegible]
by dryness the [illegible] data in columns:-

Projected area of body	[illegible] sq. ft.
Total head lost in tube	[illegible] ft.
Velocity of approach	[illegible] ft. per sec.
Mass rate of flow	[illegible] lbs. per hour

Log Pressure	1.82	1.89	1.10
Mean rate of water, lbs. per hour	2753	6000	2910
Density of water, lbs. per cu. ft.	62.3	62.3	62.3
Mean density of mixture, lbs. per cu. ft.	54.42	54.3	54.84
Density of cast mixture, lbs. per cu. ft.	71.44	62.9	72.84
Volume density in mixer, lbs. per cu. ft.	61.84	74.9	79.4
Velocity of water, ft. per sec.	5.21	4.47	4.47
Adjusted velocity in mixer, ft. per sec.	7.48	1.48	7.48
Velocity at exit, ft. per sec.	49.7	78.35	84.1
Available head, feet	48.25	47.20	72.6
Total losses, feet	23.24	29.28	17.4

Graph 30 is the equilibrium diagram for this work.


Graph No. 9
Case VI
3000 psi, 695.4°F
Losses
Head
Feet Head
21
20
19.95
19
18
0.778
0.75
0.76
0.77
0.78
0.79
Top Dryness


Graph No. 10
Case VII
3100 psi, 700.3°F
Losses
Head
Feet Head
22
21
20
19.7
19
18
1.01
1.03
1.05
1.07
1.09
1.10
Top Dryness

Summary of Equilibrium Circulation Data

For the Preceding Seven Cases

TABLE 1

Case number	I	II	III	IV	V	VI	*VII
Equilibrium top dryness	0.163	0.1515	0.1865	0.251	0.380	0.778	1.090
Steam by volume, percent	89	79	73	72	77	92	100
Mass rate of steam, lbs per hr.	533	641	770	956	1269	2155	2810
Mass rate of water, lbs. per hr.	3270	4230	4130	3810	3338	2730	2580
Density of water, lbs. per cu. ft.	50.8	46.3	42.55	38.9	34.8	28.9	26.95
True mean density in riser, lbs. per cu. ft.	14.5	21.5	23.4	23.5	22.0	19.3	18.1
Density of exit mixture, lbs. per cu. ft.	5.96	11.66	14.2	15.0	14.8	13.4	12.7
Midpoint density in riser, lbs. per cu. ft.	28.38	28.98	28.35	27.0	24.8	21.2	19.8
Velocity of water, ft. per sec.	3.28	4.66	4.95	4.99	4.95	4.82	4.88
Midpoint velocity in riser, ft. per sec.	15.64	11.6	9.9	9.0	8.3	7.6	7.6
Velocity at exit, ft. per sec.	28.0	18.5	14.9	12.9	11.7	10.4	10.4
Available head, ft.	43.2	32.2	27.1	24.0	21.9	19.95	19.7

*This case is unrealistic since it shows more steam leaving the tube than water entering. It is included here merely to show how natural circulation breaks down as the pressure approaches critical.

For the simple U-tube and the conditions of tube size and heat absorption chosen, the steam by volume is too high in the 500 psi case and satisfactory in the next four cases. At less than 3000 psi it again becomes unsatisfactory. It should be remembered that in an actual boiler circuit the friction loss would be greater due to headers and restrictions, and any heating of the downcomer would reduce the available head. These would combine to increase the top dryness and, therefore, the steam by volume so that the one inch I.D. tube would prove unsatisfactory at all pressures.

The circulation ratio in Case I. is 7.14:1

$$C.\ R. = \frac{\text{mass rate water}}{\text{mass rate steam}} = \frac{1}{\text{top dryness}} = \frac{7.14}{1}$$

This is too low for tube protection. In order to bring the steam by volume down to 75% it would be necessary to have a top dryness of 0.06 which corresponds to a circulation ratio of 17.7:1. Under these conditions a 10% reduction in water flow would reduce the circulating ratio from 17.7 to 15.9. This provides a considerable margin for inequality of flow between various tubes in a given circuit. In order to get this high circulation ratio without altering tube size, forced circulation would need to be used since there is only one equilibrium condition for these conditions in natural circulation. The use of forced circulation with its greatly increased circulation ratio to bring the steam volume in Case I down to safe limits, necessitates an increase in water and wet steam velocities in the circuit. This increases the losses, but since there is always a positive head on the circuit, they are of minor consideration. To illustrate the

calculation for forced circulation, take the data as originally computed in Case I for a top dryness = 0.06 corresponding to 75% steam by volume.

$$\dot{m} \text{ (steam)} = 533 \text{ lbs. per hr.}$$

$$\dot{m} \text{ (water)} = \frac{533}{0.06} = 8880 \text{ lbs. per hr.}$$

Area of 1 inch I. D. tube = 0.00545 sq. ft.

Density water = 50.8 lbs. per cu. ft.

Density exit = 13.47 lbs. per cu. ft.

$$\text{Velocity of water} = \frac{8880}{3600 \times .00545 \times 50.8} = 8.92 \text{ ft. per sec.}$$

$$\text{Velocity at exit} = \frac{8880}{3600 \times .00545 \times 13.47} = 33.6 \text{ ft. per sec.}$$

$$\text{Midpoint velocity in riser} = \frac{8.92 + 33.6}{2} = 21.26 \text{ ft. per sec.}$$

$$\text{Circulation Ratio} = \frac{533 + 8880}{533} = 17.7$$

At high pressure of say 2000 psi as in Case IV the natural circulation ratio for equilibrium is 5:1. Any reduction in water flow under these conditions might result in a dangerous condition. In a natural circulation unit of conventional design, it is extremely difficult, if not impossible, to be assured of uniform distribution of water to all the tubes in a given circuit. Recirculation is employed, but not all circuits operate with the same circulation ratio, because of difference in length, and variations in heat transfer due to physical location in the furnace, degree of slagging, soot formation, etc. It

$$\text{Velocity of sound} = \frac{\qquad\qquad}{\qquad\qquad} = \qquad \text{ft. per sec.}$$

$$\text{Velocity of air} = \frac{\qquad\qquad}{\qquad\qquad} = \qquad \text{ft. per sec.}$$

$$\text{Relative velocity to plane} = \frac{\qquad\qquad}{\qquad\qquad} = \qquad \text{ft. per sec.}$$

$$\text{Correction factor} = \frac{\qquad\qquad}{\qquad\qquad} = \qquad .$$

is necessary to design for average conditions corresponding to a higher circulation ratio than the minimum required. In a high pressure natural circulation unit the minimum practical ratio is 5 to 1 and most designers prefer to use 6 or 7 to 1. This results in only 60 to 65% steam by volume leaving the circuit, but gives protection to the tubes when water flow is reduced.

In a controlled forced circulation unit, orifices are installed at the entrance to the heated tubes which equalizes the water flow between various tubes and counteracts the effect of variations in heat absorption between tubes. Under these conditions it is safe to design for a lower circulating ratio such as 4:1 with a correspondingly higher percent steam by volume.

The summary of the seven calculations brings out how the density of saturated water decreases with pressure from 50.8 pounds per cubic foot at 500 psi to 28.9 pounds per cubic foot at 3000 psi. At the same time the density of the wet steam at exit increases and then falls off. This is due to the varying percentage of water in the steam. The true mean density in the riser increases as the steam volume decreases and decreases in Case VI when the steam by volume increases. These changes in density affect the available head and the velocities as is seen in Table 1.

The increase of true mean density tends to decrease the available head, while the decrease of water density also decreases the head. The available head decreases continuously throughout the range of pressure investigated. Even when the true mean density started decreasing, the decrease in water density was greater so that the available

head continued to drop, but at a lesser rate. This means that there is less head available to overcome the losses as pressure is increased. The designer of natural circulation must make every effort to keep the losses, especially friction, to a minimum particularly at higher pressures. This is the main reason why large tubes are used as will be discussed later. In forced circulation the head is supplied by the pump and does not depend on the pressure in the circuit.

The water velocity from Table 1 after an initial increase from Case I to Case II stayed nearly constant. This was because the mass rate of water decreased with increase in pressure in approximately the same ratio as the water density dropped off. The exit velocity, however, dropped off to less than one-half its initial value because the increase in exit density had a greater effect than the mass rate of water. The velocities at exit for the high pressures create another problem for the designers of natural circulation. If, due to changing load or some other happenstance, the flow is reduced in a particular circuit for a short while, the exit velocity will be lowered thus causing the tube wall to heat up. At high pressures the saturation temperatures and, therefore, the tube wall temperatures are already quite high. Such slowing of the cooling mixture in the tube could easily cause overheating of the tube metal and consequent failure especially if repeated very often. The same thing could happen in any sluggish circuit if, for example, a piece of slag were to drop off exposing the tube to a greater heat input locally. The steam formed would be held back by the slower moving mass in the tube above and the tube metal would tend to overheat. This cannot happen

in forced circulation because of the flow regulating orifices at the entrance to all tubes and the constant positive head on the system.

It has been shown how Case I may be corrected by applying forced circulation. The designer of natural circulation can also correct this design to insure sufficient water for tube protection. There are two methods which may be used, namely: (1) reducing the tube length and (2) increasing the tube diameter.

(1) Reducing the tube length is undesirable since this reduces the projected area of tube exposed to the heat of the furnace, which, in turn, reduces the amount of steam generated and, therefore, the steaming rate of the boiler. If the tube length in Case I is reduced from 60 feet to 30 feet, the mass rate of steam is reduced from 533 pounds per hour to 266 pounds per hour. The steam by volume has now been reduced to 73 percent which means there is enough water present to protect the tube but only at the expense of steam generation.

(2) Increasing the tube diameter is the most likely way of correcting this design. Suppose the inside tube diameter is increased from one inch to two inches, the mass rate of steam is thereby increased from 533 pounds per hour to 1030 pounds per hour which means nearly twice as much working fluid, and the steam by volume is now only 64.5 percent.

In an actual boiler circuit the steam by volume would, of course be higher than for this ideal circuit, but unless of unusual design, the tube would still be protected with the volume of steam probably not exceeding 70 percent. In practice, however, boiler tubes in boilers designed for around 500 psi will range from $2\frac{1}{4}$ to over $3\frac{1}{2}$ inches I.D.

with the latter being the more common. This is because of lack of faith in the various theories of circulation. Some of the theories in themselves are fairly good, such as the one illustrated, but with most types of boilers, it is impractical to attempt any rigorous analysis of circulation, due to the indeterminate division of flow between tubes connected in parallel between drums and headers. As mentioned before some circuits are exposed to greater heat input than others because of their physical location within the boiler. The degree of slagging, sooting, scale formation, and type of fuel fired, all cause local variations in heat absorption in the tubes. The designer must design the boiler so that even the circuit with the highest heat absorption is protected at all loads. This means that the majority of the circuits circulate more water than necessary for their protection alone. With so many indeterminate factors the boiler designer must use a large factor of safety represented by the large tubes above.

At pressures of 800 to 1500 psi, modern boilers use somewhat smaller tubes. This tendency is substantiated by Table 1 which shows the least steam by volume to be present for Case III (1000 psi) and Case IV (1500 psi). On the other hand a return to four inch O.D. tube size (3.52 inch I.D.) was deemed necessary in the design of a recent high capacity natural circulation boiler operating at the unusually high pressure of 2450 psi. Table 1 also substantiates this, since Case V (2500 psi) and Case VI (3000 psi) show an increase in volume of steam over the previous case, indicating a need for larger diameter tubes.

As was shown in the calculations the head available for circulation is the head created by the difference in density of the fluid in the downflow and upflow circuits, minus the loss in head caused by fluid friction, turbulence and acceleration. If too much head is used in fluid friction and turbulence, then too little remains available for circulation. These facts impose limits on the arrangement of steam-generating tubes and also result in large tubes which have less friction loss than a small tube. To illustrate using the data from the calculations for 500 psi:-

1 inch I.D. tube

Friction loss in downcomer

$$\frac{4fZU_s^2}{d \times 2g} \times \text{rel. den.} = \frac{4(.006)(60)(5.57)^2}{.0833 \times 64.35} \times 1 = 8.35 \text{ feet}$$

Friction loss in riser

$$\frac{4(.006)(60)(14.56)^2}{.0833 \times 64.35} \times .383 = 21.88 \text{ feet}$$

2 inch I.D. tube

Friction loss in downcomer

$$\frac{4(.006)(60)(7.80)^2}{.167 \times 64.35} \times 1 = 8.28 \text{ feet}$$

Friction loss in riser

$$\frac{4(.006)(60)(12.4)^2}{.167 \times 64.35} \times .591 = 12.2 \text{ feet}$$

In forced circulation, tube friction does not detract from the circulation head and is, therefore, a minor consideration. This helps to explain why small tubes and higher velocities can be used.

B. Forced Circulation Problems

In forced circulation the designer has much more control over the various factors involved in circulation since there is at all times a positive head on the system which emanates from the pump. Using the previously calculated example of Case I involving forced circulation, page 71, the pertinent data is:

$$\dot{m} \text{ (steam)} = 533 \text{ lbs. per hr.}$$

$$\dot{m} \text{ (water)} = 8880 \text{ lbs. per hr.}$$

The circulating pump must handle 533 + 8880 = 9413 pounds of water per hour. In this case a large excess of water flows through the circuit which indicates a forced circulation boiler of the recirculation type.

In this type boiler the net positive head on the system assures flow in the right direction at all times in the tubes. The circulating pump which is the key to the whole design presents several interesting problems. The temperature at the pump is equal to the saturation temperature for drum pressure. The pressure at the pump is higher than drum pressure by the pressure equivalent of the difference in elevation, minus the head loss in the piping since the pump must always be placed at a lower level than the surface of the liquid in the boiler. If the boiler pressure should drop suddenly

by a few pounds, as sometimes happens, there is danger that the circulating pump will flash because the water at the pump suction is still at the temperature of saturation of former drum pressure. This tendency to flash decreases with increase in pressure. At 400 psi boiler pressure a drop of $1^{\circ}F$ means only 3.8 psi decrease, but at 1200 psi, it means 9 psi decrease. It is highly desirable, therefore, that this pump operate with a considerable net positive suction head. This can be assured by designing so that the piping loss is substantially less than the pressure increase due to elevation, and by "stabilizing" the circuit. "Stabilizing" means to inject the make up feed water directly into the circulating pump suction instead of the economizer. If the make up feed water is at $300^{\circ}F$, this is $234^{\circ}F$ below the saturation temperature of $534^{\circ}F$ corresponding to 900 psi. The quantity of this feed water is small, but it is sufficient to lower the pump temperature several degrees thus preventing flashing.

The high pressures and temperatures of the pumps create severe stuffing box problems. Many ingenious designs have been worked out, but this feature is subject to improvement, especially at very high temperatures. By using an overhung impeller with both bearings on the same side of the pump, only one seal is required. Figure 17 illustrates this type which is preferred in this country. Some foreign installations contain a bearing and a seal on each side of the pump. Spare circulating pumps are usually installed to take care of pump breakdowns which have been surprisingly few in practice, and to allow pump inspection and repacking without reducing load on the boiler.

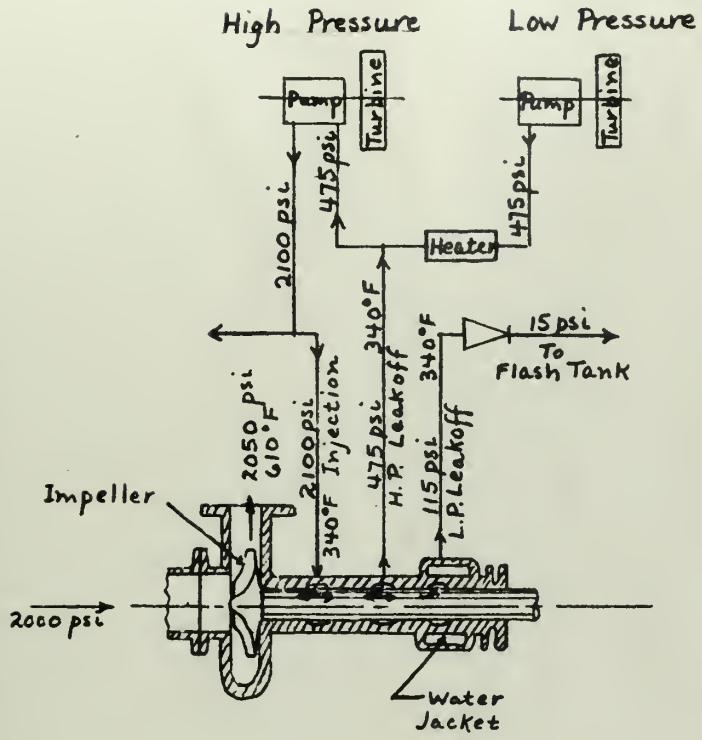


Fig 17 Typical schematic arrangement of a Forced Recireulation boiler circulating pump

These pumps are usually electric with one or both also equipped to be driven by a steam turbine in case of power failure.

Circulating pumps are designed to handle as a rule five to ten times the maximum boiler feeding capacity. Consequently, they are larger and slower than feed pumps, and also more costly. The gross power consumption is of the order of 0.5 to 0.6 percent of the boiler output. This is not all a loss because forced circulation permits the use of such economical heating-surface arrangements that fewer baffles are required, and the power used by the pumps may often be offset by low power consumption of the induced-draft fans. Some energy is also returned to the water as heat because of the increase in pressure.

The orifices used in forced circulation steam generators are installed at circuit entrances and are of such size that they regulate the flow of water to each circuit according to its length, arrangement, and heat absorption. When small diameter orifices are used, it is necessary to provide strainers between the circulating pump discharge and the orifices, to prevent foreign matter from plugging the orifices at the entrance to the generating circuit. It is not necessary to have a different size orifice for each circuit. Practice has shown that two or three sizes will suffice for all circuits in a boiler.

On the basis of circulation alone then, natural circulation problems center around the vagaries and uncertainties in flow and flow distribution through the system, with its consequent need for

large tubes to insure tube protection. In forced recirculation the flow is always positive and properly distributed. The problems center around the pump design and orifice and strainer details. Small tubes are characteristic of forced flow.

In the case of "once through" boilers circulation as such does not enter the picture, since it is only necessary for the feed pump to supply an amount of water sufficient to generate the required amount of steam out the other end of the boiler. In this case salt deposition and the control of the boiler are the big problems.

2. The Advantages of Small Tubes

Large diameter tubes are characteristic of natural circulation boilers as explained in the previous section on circulation. In the same way small diameter tubes are typical of forced circulation. The smaller tubes have a number of decided advantages especially at high pressures. This is one of the inherent advantages of forced circulation.

The large tubes operate at considerably higher metal temperatures than the small ones when the temperature of the saturated water is the same in both cases. This places the large tube at a disadvantage, since thermal stresses are important in boiler tube design. To illustrate this phenomenon, the mean metal temperatures and the temperature gradients for tubes of 1.15 inches and 4 inches outside diameters will be computed. (See Appendix "D")

The heat transfer coefficient h_i between the inside tube surface and the saturated water may be calculated from McAdams formula:

$$\frac{h_i D_i}{k} = 0.023 \left(\frac{D_i U d}{v}\right)^{0.8} \left(\frac{cp\ v}{k}\right)^{0.4}$$

where D_i denotes the inside diameter of the tube, k the termal conductivity, U the velocity, d the density, v the viscosity and cp the specific heat.

The temperature drop from the inside tube wall to the saturated water flowing in the tube is a conduction process across the inside film and is given by Isaac Newton's formula

$$\frac{hD}{k} = 0.023 \left(\frac{\rho v D}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$$

$$q = h_i A (t_s - t)$$

where q is the rate of heat flow, A is area at right angles to the flow, t_s is inside tube wall temperature, and t is temperature of the saturated water.

The temperature gradient across the tube wall is obtained from the conduction equation particularized for a curved tube:

$$q = k_m A_m \frac{(t_o - t_s)}{(r_o - r_i)}$$

where k_m is the mean thermal conductivity, A_m is the logarithmic mean area, t_o and r_o are outside temperature and radius respectively, t_s is same as above, and r_i is the inside tube radius.

As in the circulation calculations a constant rate of heat absorption equal to 70,000 BTUs per square foot per hour and a tube height of sixty feet will be assumed. The pressures and corresponding saturated water temperature are indicated on the sample set of calculations.

$$\dot{q} = \frac{k}{L}(T_2 - T_1)$$

where k is the mean thermal conductivity, L is the thickness of the material, and T_1 and T_2 are the inside temperature and outside temperature, respectively.

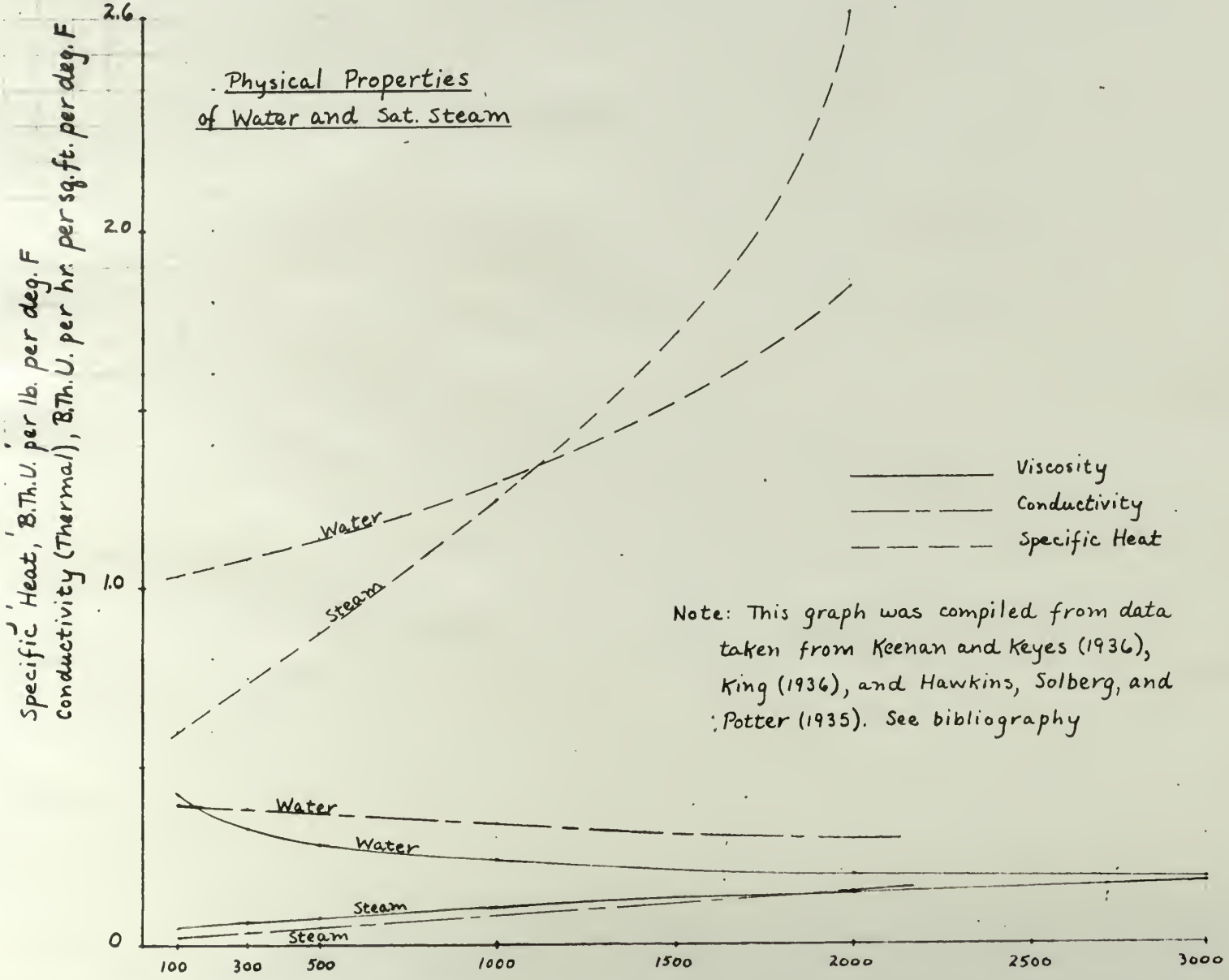
The solution of this equation is extremely easy if heat is assumed to flow only in one direction. The geometry and temperature distribution of many practical problems are obtained as the sum or difference of solutions.

The following equation is obtained for a plane slab:

$$\dot{q} = -k \frac{(T_2 - T_1)}{(x_2 - x_1)}$$

where k is the mean thermal conductivity, T_2 is the temperature at x_2, and T_1 is the temperature at x_1. The negative sign indicates that heat flows in the direction of decreasing temperature.

Graph No. 11


Physical Properties
of Water and Sat. Steam
Specific Heat, B.Th.U. per lb. per deg. F
Conductivity (Thermal), B.Th.U. per hr. per sq. ft. per deg. F
2.6
2.0
1.0
0
Water
Steam
Viscosity
Conductivity
Specific Heat
Note: This graph was compiled from data taken from Keenan and Keyes (1936), King (1936), and Hawkins, Solberg, and Potter (1935). See bibliography
Water
Water
Steam
Steam
100
300
500
1000
1500
2000
2500
3000
Pressure, lbs. per sq. in.

Calculation of Temperatures and Temperature Gradients

at Entrance or Water End of a 1.15 inch O. D. Tube

Pressure 1000 psi

Saturation Temperature 544.6°F

From Graph 11 and Appendix "D" the following values are obtained:

k = 0.338 BTU per hr. ft.° F

D_i = 0.95 inch or 0.0793 feet

U = 4.44 ft. per sec. or 16000 ft. per hr.

d = 46.3 lbs. per cu. ft.

v = 0.235 lbs. per hr. ft.

cp = 1.29 BTUs per lb.° F

Now substitute the values in McAdams formula:

$$h_i = \frac{k}{D_i} \times 0.023 \times \left(\frac{D_i \, V \, S}{v}\right)^{0.8} \times \left(\frac{cp \times v}{k}\right)^{0.4}$$

$$h_i = \frac{0.338}{0.0793} \times .023 \times \left(\frac{0.0793 \times 16000 \times 46.3}{0.235}\right)^{0.8} \times \left(\frac{1.29 \times 0.235}{0.338}\right)^{0.4}$$

h_i = 1993 BTUs per hr. sq. ft. °F.

With this value of the inside heat transfer coefficient the temperature drop from the inside tube surface to the flowing water is calculated and since the water temperature t = 544.6° is known, the inside tube temperature t_s may be obtained. Let A = 1 sq. ft.

Calculation of Temperature and Temperature Gradients

at Interface of Steel and Lead Wool Insulation

Pressure 1200 psi

Saturation Temperature 314.4°F

From Curve 11 and Appendix VIII the following values are obtained:

k = 0.296 BTU per hr. ft.^{2} F

c_p = 0.15 Btu per lb. 0.070 Pmol

μ = 4.44 lb. per sec. or 1604 lb. per hr.

d = 4.63 lbs. per cu. ft.

r = 0.291 Btu. per lb. F.

cp = 1.00 BTUs per lb.^{o} F

Now substitute the values in Nielsen formula:

$$Nu = \frac{hd}{k} = 0.023 \times \left(\frac{dv\,d\,c_p}{k}\right)^{0.8} \times \left(\frac{c_p\,\mu}{k}\right)^{0.4}$$

$$h = \frac{0.296}{0.0139} \times .023 \times \left(\frac{0.0139 \times 1604 \times 1.00}{0.296}\right)^{0.8} \times \left(\frac{1.00 \times 0.296}{0.296}\right)^{0.4}$$

h_i = 1900 BTUs per hr. sq. ft. ^oF.

With this value of the inside heat transfer coefficient the temperature drop from the inside tube surface to the flowing water is calculated and minus the water temperature t = 314.6^o is known, the inside tube temperature t_s may be obtained. Let A = 1 sq. ft.

$$q = h_i A (t_s - t)$$

$$t_s = \frac{q}{h_i A} + t$$

$$t_s = \frac{70000}{1993 \times 1} \times 544.6^\circ$$

$$t_s = 579.7^\circ F$$

To get the temperature gradient across the tube thickness, a value of k_m must be known. This property is a function of temperature and type of steel. Several types of steel are currently popular for tube manufacture, such as carbon steel, one half molybdenum - carbon steel, carbon - silicon steel, etc. These all have approximately the same thermal conductivities. Appendix "C" is a table of the values of k for a typical carbon (C = 0.5%) steel. Interpolation in this table will give the desired value of k. The inner tube surface temperature, t_s, being the only tube temperature known so far, will be used to determine an approximate k_m. The method of successive approximations may be used to refine this k_m, but since the difference is quite small it will be neglected here. From Graph 1 the tube thickness = 0.10 inch.

Therefore $r_o = 0.575$ inch or 0.048 ft.

$r_i = 0.475$ inch or 0.0396 ft.

For $t_s = 579.7^\circ$; $k_m = 25.94$ BTUs per hr ft $^\circ F$

Let $A_m = 1$ sq. ft.

$$t_f = 0.11\,(t_s - t_i)$$

$$\tau_s = \frac{\rho c}{k}\, r^2$$

$$t_s = \frac{2q\sqrt{\tau}}{k\sqrt{\pi}} + 2t_i - t_o$$

$$\tau_s = 27.4\,r^2$$

To get the temperature reached across the tube thickness, t value of t_f must be used. This property is a function of temperature and type of steel. Several types of steel are currently employed for tube manufacture, such as carbon steel, low alloy, medium carbon steel, nickel - chrome steel, stainless steel, etc. These all have approximately the same thermal conductivities. Appendix VIII is a table of the values of k for a typical carbon ($c = 0.21$) steel. Interpolation in this table will give the desired value of t_f. The tube surface temperature, t_s, being the only time temperature known, may be used to determine an approximate k. The method of successive approximations may be used to refine final t_f, but since the difference is quite small it will be neglected here. Also since a tube thickness = 0.19 inch.

Therefore:

for t_f = 571.7°; k = 25.0 Btu per hr. ft. °F

let t_f = 1 sec. ft.

$$q = k_m a_m \frac{(t_o - t_s)}{(r_o - r_i)}$$

$$t_o = \frac{q(r_o - r_i)}{k_m A_m} + t_s$$

$$t_o = \frac{70000(.048 - .0396)}{25.94 \times 1} + 579.7$$

$$t_o = 602.2^{\circ}F$$

The arithmetic mean temperature in the tube wall at the lower end is:

$$t_m = \frac{t_o + t_s}{2} = \frac{602.2^{\circ} + 579.7^{\circ}}{2}$$

$$t_m = 687.5^{\circ}F$$

The temperature gradient across the tube wall is the difference of t_o and t_s

$$\text{Temp. Grad.} = 602.2^{\circ} - 579.7^{\circ} = 22.5^{\circ}F$$

As the pressure and corresponding saturated temperature of water increase the effect on the metal temperature and gradient becomes more pronounced. Table 2 below summarizes the data for the 1000 psi calculation just completed and also 1500 psi and 2000 psi. See Appendix "D" for equilibrium data.

TABLE 2

Pressure, psi	1000		1500		2000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Diameter, I.D. inches	0.95	3.48	0.89	3.34	0.85	3.18
h_i, BTUs per hr sq ft $^{\circ}$F	1993	2055	2130	2225	2060	1896
Inside Tube Temp(t_s), $^{\circ}$F	579.7	578.7	629.2	627.7	669.8	672.8
Outside Tube Temp(t_o), $^{\circ}$F	602.2	638.1	659	705	705.2	764.8
Mean Tube Temp. (t_m), $^{\circ}$F	591.1	608.4	644.1	666.3	687.5	721.3
Temp. Grad. $^{\circ}$F	22.5	59.4	29.8	77.3	35.4	97.0
Saturation Temp. $^{\circ}$F	544.6	544.6	596.2	596.2	635.8	635.8

The above table shows how the larger tubes have a higher mean temperature which becomes increasingly higher at a slow rate as the pressure increases. The difference in temperature gradient across the tube metal is very pronounced, the gradient being nearly three times as great for the large tube. The ratio is slightly less for the 1500 psi case. This is attributed to the fact that in plotting Graph 11, data was available for points at 1000 psi and 2000 psi whereas the properties at 1500 psi were taken from the curves as drawn through all the points.

The temperature of the metal at the exit of the tubes cannot be ascertained with any degree of certainty. Instead of a low viscosity liquid there is now a mixture of dry steam and water. No data are available regarding the heat transfer to wet steam. It can hardly be less than dry steam, and in all probability is more, due to the presence of water particles in the steam.

Professor Einstein has developed an equation for the viscosity of hard spheres in a liquid

$$v = v_o(1 + 2.5\phi)$$

v_o is viscosity of the solvent

ϕ is total volume of spheres in 1 cu. ft.

An attempt was made to utilize this formula to get the viscosity of wet steam by altering the 2.5, but any such constant would need to be a variable. The conductivity and specific heat were then estimated by taking volume and weight percentages respectively of the steam and water. These values were then substituted into McAdams formula, but the results were disappointing. Such use of Einstein's equation does not account for such variables as the large density difference between liquids as a solvent and dry steam as a solvent plus the added compressibility of the dry steam. There are other unknowns such as the applicability of McAdams formula to mixtures. The computation of the conductivity was a guess, no real basis exists for the method attempted except that volume percentages were used since conductivity is a function of a linear dimension. Investigation of any of these properties would be an excellent laboratory research project and is so recommended for thesis work in the future.

In order to get an idea of tube metal temperatures and gradients at the exit of these tubes, dry steam values of viscosity, conductivity, and specific heat will be used, even though these are all probably on the low side. Other data is as previously calculated for the exit conditions of the 1.15 inch and 4.0 inch tubes. See Appendix "D".

TABLE 3

Pressure, psi	1000		1500		2000	
Diameter, O.D., inches	1.15	4.0	1.15	4.0	1.15	4.0
Diameter, I.D., inches	0.95	3.48	0.89	3.34	0.85	3.18
h_i, BTUs per hr sq ft°F	1138	1156	1310	1451	1775	1751
Inside Tube Temp(t_s),°F	606.1	605.2	649.7	644.5	675.3	675.8
Outside Tube Temp(t_o),°F	628.9	665.3	679.8	722.3	710.8	773
Mean Tube Temp. (t_m), °F	617.5	635.2	664.7	683.4	693	724.4
Temp. Grad.,°F	22.8	60.1	30.1	77.8	35.5	97.2
Saturation Temp.,°F	544.6	544.6	596.2	596.2	635.8	635.8

It should be noted that while the temperature gradient scarcely varies along the tube, the mean temperature rises at a decreasing rate. Table 4 is the arithmetic mean average of Tables 2 and 3.

TABLE 4

Pressure, psi	1000		1500		2000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Mean Tube Temp(t_m),°F	604.3	621.8	654.4	674.9	690.3	722.9
Temp. Grad.,°F	22.7	59.8	30.0	77.6	35.5	97.1

In the above calculations a constant rate of heat absorption of 70,000 BTUs per square foot per hour was used. An increase in this rate will result in a corresponding increase in temperature gradient, mean tube temperature, and outside tube temperature. Table 5 is a comparison

of these factors at 2000 psi for heat absorption rates of 70,000, 100,000, and 140,000 BTUs per square foot per hour. These are all taken at the entrance or water end of the riser.

TABLE 5
(2000 psi, 635.8°F)

Heat Absorption Rate BTUs per sq. ft. per hour	70,000		100,000		140,000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
h_i, BTUs per hr.sq.ft.°F	2060	1896	1952	2162	1645	2762
Inside Tube Temp., °F	669.8	672.8	688	682	720.9	686.5
Outside Tube Temp., °F	705.2	764.8	739	821.3	793.3	881.8
Mean Tube Temp., °F	687.5	721.3	714.5	751.6	757.1	784.1
Temp. Grad., °F	35.4	97.0	51	139.3	72.4	195.3

At low heat rates of 50,000 - 70,000 BTUs per square foot per hour the large tubes will not exceed the A.S.M.E. mean temperature limits even though these are considerably in excess of the small tube mean temperatures. A lengthy series of experiments indicated heat transfer rates in front-row boiler tubes reached 143,000 BTUs per square foot per hour or higher. Table 5 shows that for such a heat rate, if at high pressure, the large 4.0 inch tube would be unsuitable because of excessive tube metal temperature. At this pressure even a heat rate of 100,000 BTUs per square foot per hour would be questionable for such a tube, since the outside metal temperature would be over 820°F. The smaller tube would be quite satisfactory at this heat rate, and could be made to work at 140,000 BTUs per square foot per hour if the pressure is dropped slightly.

The tube thicknesses as used by the A.S.M.E. Power Boiler Code are based on the working pressure stress only with a 750°F. mean maximum metal temperature (See Graph 1). Except possibly at very low pressures, the wall thicknesses of boiler tubes are always greater than one-tenth the radius. The thick wall theory of stresses is, therefore, applied. Appendix "E" gives a development of this theory from which it is determined that the maximum tensile stress occurs at the inside tube surface and is equal to:-

$$\sigma_{\theta_{MAX.}} = \frac{-p_i (a^2 + b^2)}{a^2 - b^2}$$

At 1000 psi for the case of the 1.15 O.D. inch tube $a = 0.475$ inch and $b = 0.575$ inch

$$\sigma_{\theta_{MAX.}} = - \frac{1000 (0.475^2 + 0.575^2)}{0.475^2 - 0.575^2} = - \frac{1000 (.5561)}{- .1051}$$

$$\sigma_{\theta_{MAX.}} = - \frac{556.1}{.1051} = 5300 \text{ psi}$$

For the several pressures previously used the maximum tensile stresses are tabulated in the table below for the 1.15 and 4.0 inch tubes.

TABLE 6
Maximum Tensile Stresses Due to Pressure Only

Saturation Pressure, psi	1000		1500		2000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Outside Radius, b, inches	0.575	2.0	0.575	2.0	0.575	2.0
Inside Radius, a, inches	0.475	1.76	0.445	1.67	0.425	1.59
Max. Tensile Stress, psi	5300	7250	5960	8410	6830	8890

In addition to the stress due to the pressure of the water and steam, there is also a thermal stress which is a function of the temperature gradient. A formula has been developed which gives the maximum thermal stress in a thick walled cylinder. This stress is also a maximum in a boiler tube at the inside surface.

$$\sigma_{\theta\,(r=a)\,max} = \frac{E\ (t_s - t_o)}{2(1-V)\ \ln \frac{b}{a}} \left[1 - \frac{2b^2}{b^2 - a^2} \ln \frac{b}{a}\right]$$

where for steel

$$E = 30 \times 10^6 \text{ psi}$$

$$\alpha = 70 \times 10^{-7}$$

$$\nu = 0.3$$

Using the conditions of the 1.15 O.D. inch tube at 1000 psi saturation pressure this becomes:-

$$\sigma_\theta = \frac{(30 \times 10^6)(70 \times 10^7)(-22.5)}{2(1 - 0.3) \ln \frac{.575}{.475}} \left[1 - \frac{2(.575)^2}{(.575)^2 - (.475)^2} \ln \frac{.575}{.475}\right]$$

$$\sigma_\theta = \neq 3540 \text{ psi}$$

Table 7 is a tabulation of these thermal stresses for the previously used saturated pressure conditions of the 1.15 and 4.0 inch tubes.

TABLE 7
Maximum Tensile Stress Due to Temperature Alone

Saturation Pressure, psi	1000		1500		2000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Outside Radius, b, inches	0.575	2.0	0.575	2.0	0.575	2.0
Inside Radius, a, inches	0.475	1.76	0.445	1.67	0.425	1.59
Max Thermal Stress, ,psi	3540	11440	4840	13250	5960	16470

From this table when compared with Table 6 it can be seen that the thermal stresses in large tubes are larger than the pressure stresses. They are also much larger, approximately 3:1, than the thermal stresses in the small tubes. The total stress is the sum of the pressure and thermal stresses which is given in Table 8.

TABLE 8

Saturation Pressure, psi	1000		1500		2000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Total Stress, psi	8,840	18,690	10,800	21,660	12,790	25,360

Table 7 is a tabulation of these thermal stresses for the previously used geometry over the conditions of the 2.33 mil 4.0 Mev beam.

TABLE 7

Maximum Tensile Stress due to Temperature Rise

Material	300	1200	2400
Aluminum, 6061 Alloy	1,035 psi	1,035 psi	1,035 psi
Molybdenum Coating	2,270 psi	2,270 psi	2,270 psi
Double Coating	1,475 psi	1,475 psi	1,475 psi
Net Thermal Stress	33,440 psi	33,440 psi	33,440 psi

These data should be compared with Table 6 to see to what extent the thermal stresses in these cases are larger than the pressure stresses. They are also much larger, approximately 1:5, than the thermal stresses in the small beam. The effect of the rise of the maximum thermal stresses with is given in Table 8.

TABLE 8

Material	300	1200	2400
Aluminum, 6061 Alloy	1,035 psi	1,035 psi	1,035 psi
Total Stress, psi	34,475 psi	34,475 psi	34,475 psi

TABLE 9
(2000 psi, 635.8°)

Heat Absorption Rate BTUs per sq ft. per hour	70,000		100,000		140,000	
Diameter, O.D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Thermal Stress, psi	5,960	16,450	8,600	23,600	12,200	33,100
Total Stress, psi	12,790	25,360	15,430	32,490	19,030	41,990

The factor of safety in a boiler tube selected according to A.S.M.E. Code is based on pressure stress alone. From Table 8 it is observed that the total stress increases markedly with increase in pressure. Increasing the rate of heat transfer causes an even more sharp rise in total stress. As this total actual stress increases the factor of safety decreases, so that the actual factor of safety may be very low with large thick wall tubes operating at high heat transfer rates and high pressures.

The yield point and ultimate limit of the steels currently in use for boiler tubes varies considerably. As a rule the yield point varies from 35,000 to 45,000 psi and the ultimate limit from 55,000 to 65,000 psi. From Table 8 it is seen that the total stresses are in all cases below the yield point. At 2000 psi, however, the factor of safety defined as $n = \sigma_{yp.} / \sigma_{working}$ is less than 2 for the large tube whereas it is about 3 for the small tube. As the heat rate is increased as in Table 9 the factor of safety has dropped to unity in the 140,000 BTUs per square foot per hour case for the 4.0 tube, while the 1.15 inch tube has a factor of about 2. This heat rate is not uncommon nowadays and higher ones are being recorded in certain circuits of modern boilers. Pressures above

2000 psi are appearing more frequently in the newer boilers. Unless the heat absorption rate is kept down the total stress on large tubes may exceed the yield point and approach the ultimate limit, even with adequate circulation. This is a serious limitation in natural circulation design. The small diameter thin-wall tubes used in forced circulation provide a high safety factor because of lower outside and mean tube wall temperatures and much lower temperature gradients. This factor of safety exists even at very high heat absorption rates.

In the case of the recently installed unit of the Etiwanda Station of Southern California Edison Company which is designed for 2100 psi, the 1-1/4 inch, 1-1/2 inch and 1-5/8 inch O.D. boiler tubes of the controlled circulation unit are suitable for an operating pressure of 2,332 psi, compared with 2,010 psi for 3 inch O.D. tubes normally furnished with 1800 psi natural circulation boilers.

It is of interest to compare the weight of tube metal and of water as between a 1.0 inch O.D. and a 4.0 inch O.D. boiler tube. The weight of steel is about 490 pounds per cubic foot. Let the tube length be 60 feet.

<u>For the one inch Tube at 1000 psi sat.</u>

$$\text{Weight} = \frac{\pi}{4} (D_o^2 - D_i^2) \times 60 \times 490$$

Weight = 63.3 pounds

<u>For the four inch Tube at 1000 psi sat.</u>

Weight = 620 pounds

A furnace wall made up of one inch tubes would contain four times as many tubes as a four inch wall. However, four one-inch tubes weigh only 4 x 63.3 = 253.2 pounds as compared to the 620 pounds of the one four inch tube. This means a saving in weight of tubes alone of 60%. Because of this great weight reduction and elimination of mud drums, etc., the pressure parts of a 900,000 pound per hour, 1800 psi reheat forced circulation boiler weigh approximately 350,000 pounds (175 tons) less than a comparable natural circulation unit. This lower weight results in a substantial savings in cost of alloy and structural steel both of which are on the critical material list at the present time. The percentage saving in weight remains nearly the same as pressures and tube thicknesses vary.

The internal area of a one inch O.D. tube is only about 5.5% of that of a four inch O.D. tube.

For the one inch tube at 1000 psi

$$\text{Area (internal)} = \frac{\pi}{4} D_i^2 = .785 \times (0.82)^2 = .528 \text{ sq. in.}$$

For the four inch tube at 1000 psi

$$\text{Area (internal)} = .785 \times (3.48)^2 = 9.51 \text{ sq. in.}$$

The four one inch O.D. tubes present the same projected area exposed to the furnace heat as the one four inch tube, so the steam generated is the same in both cases. If the circulation ratio is the same then the four small tubes must receive the same amount of water as the single large tube. The internal area of four small tubes is only 4 x 5.5% = 22% that of the large tube. The velocities, therefore, at any point

in the small tubes would be greater than in the large by a factor of 100/22 = 4.55. Since such a high velocity is rarely ever needed the circulation ratio and amount of water to be circulated can be substantially reduced in small tubes. This lower circulation ratio will, however, cause the steam and water mixture velocity to increase more rapidly toward the exit.

The weight of water in the four small tubes is only 22% that of the large tube. This means there is less water to be heated when starting the boiler. This factor causes an appreciable saving in fuel and time when starting up a forced circulation unit.

The small tubes can be more easily handled and bent to fit around burners, soot blowers, etc. They can be more quickly and easily butt-welded because the amount of deposited metal for four one inch tubes is only one-third as great as for the four inch tube. The smaller tubes are much more readily fitted into place.

It is easier to clean and remove slag from a furnace wall of small tubes because they present a more nearly plane surface. This factor also tends to simplify the wall backing and insulation problems on the casing side.

Small-diameter tubes have one disadvantage in that they impart very little structural strength to a wall. This necessitates closer spacing of the structural or girth members.

3. Miscellaneous Comparisons

In addition to circulation differences and differences in tube sizes, there are a number of other basic comparisons which should be made between natural and forced circulation boilers.

The supports for the superheater, economizer, and reheater in a natural circulation boiler are exposed to very high temperatures. They are usually made of expensive, critical alloys such as 25 chrome - 12 nickel steel. Under these high temperature conditions, fuel oils having metallic salts, such as vanadium, cause serious deterioration of these supports and spacers. One solution to this problem is to water-cool the supports. In a natural circulation boiler this requires an outside source of water, with consequent loss of heat and thermal efficiency. Controlled recirculation makes it possible to provide water-cooled supports as part of the steam generating system since the positive flow of forced circulation permits almost any convenient arrangement of tubes and circuits.

Another interesting point of comparison is with respect to physical size. With many boilers now being installed out of doors the smaller volume of the forced circulation unit effects no appreciable saving in building volume in these circumstances. Where boilers are installed indoors, forced circulation with its flexibility of tube arrangement and smaller overall volume permit installation in buildings designed previously for older, low capacity units, or in other words, much greater steam output can be realized from a given building space by use of high pressure forced circulation as compared to natural circulation designs.

To illustrate some comparative figures on the physical size of two typical 1,000,000 pounds per hour capacity, 2000 psi, reheat boilers, one natural and the other forced circulation, are given. The forced circulation boiler occupies a greater floor space, 100 ft. x 60 ft. as compared to 80 ft. x 60 ft., but of much greater importance is the fact that natural circulation requires 30% greater vertical space, 102 feet as compared to 72 feet for forced circulation. Where the boiler must be enclosed this additional thirty feet in the vertical direction creates quite a drastic increase in cost of building materials and in construction costs. The installation at the Somerset, Massachusetts plant of the Montaup Electric Company is typical of this situation. The severe winter conditions in this area made housing for the boiler mandatory. The installation called for the utilization of existing space designed years before for a 200,000 pounds per hour boiler of radically different design. Remodeling the building was considered too expensive and not enough head room was available to insure satisfactory operation of a natural circulation boiler of the capacity, 650,000 pounds per hour desired. As a result the first large forced recirculation boiler in this country was installed. After a few minor initial adjustments, this boiler has given very satisfactory service for the past ten years.

The lower water-steam ratio allows all risers to enter the drum generally in two rows, which permits welded nozzles and reduced drum thickness with consequent saving in drum weight. This coupled with the elimination of one or more drums gives a substantial savings in metal needed in the boiler. This weight reduction along with the reduction

in tube weight, results in a weight savings of nearly one-half in the pressure parts over a natural circulation unit of like capacity. All pressure joints are welded and the absence of rolled joints avoids the danger of acid or caustic attack at these locations. The elimination of handhole caps and gaskets is an additional advantage. "Once through" boilers, with no drums whatsoever, show an even greater weight saving.

A further reduction in weight results from the smaller operating water content - about one-third or less that of conventional boilers.

The reduction in weight of pressure parts and water content coupled with the lower height requirements, result in a substantial savings in the cost of structural steel and foundations.

Feed water requirements differ markedly among the various types of forced circulation boilers. In the case of the LaMont type the requirements are about the same as for natural circulation. There is less tendency to scale formation, however, because of the positive circulation provided. In the Loeffler boiler there is no scale formation in any of the tubing so that water purity is of very minor importance. The other recirculation types lie between these two in their feedwater requirements, but in general more like the LaMont type. The "Once Through" type boilers on the other hand must have the purest possible type of feedwater to keep down the tube deposits. These requirements are much more stringent than in natural circulation.

One notable advantage accruing from the use of forced circulation is the facility with which clean dry steam can be obtained. In the La Mont type the circulating ratio is low, so that the percentage of steam to water entering the drum separators is much higher. Reduction of the

water loading in steam separators reduces the solids carryover with accompanying higher efficiency and capability of the turbine unit. The use of steam as a carrying agent in the Loeffler boiler gives steam of about the same dryness as in natural circulation. The Velox and Steamotive units achieve their steam dryness by virtue of their highly efficient tangential steam separators. As previously noted in the Schmidt-Hartmann boiler, because of the quiet nature of the evaporation, the moisture content of the steam does not exceed 0.01 percent. In the case of "once through" types the long tubes with evaporation early in the circuit result in steam dryness similar to the other forced circulation types. Where water injection is used for control as described in Chapter II - Sulzer the steam dryness becomes more nearly like natural circulation. In all cases of forced circulation, priming is either completely eliminated or reduced to only a possibility.

Another point of comparison is in the matter of starting up the boiler. The starting of a natural circulation unit is a rather slow process since it is necessary to heat up a large amount of water to a fairly high temperature before the natural circulation process will start. The boiler elements heat progressively which means additional time in order not to overstress some local section of the boiler. In the forced recirculation type boilers the positive circulation assures a rapid and uniform heating of the entire boiler in a short period of time. The LaMont and Schmidt-Hartmann boilers which have the highest water content of any of the forced circulation units can be safely brought on to the steam range, even in the largest sizes and highest pressures, in a matter of two and one-half hours. In the case of the

other types of boilers this time is even less - as examples; a Loeffler boiler of 139,221 pounds per hour capacity at 1840 psi and 925°F took one hour fifty-six minutes from cold until cut onto the line, a Ramzin boiler of 440,000 pounds per hour capacity at 2,840 psi and 932°F had a starting time of two hours originally, but this was later decreased to twenty-eight minutes, the Steamotive requires about ten minutes, and the Velox type has a starting time of only three to eight minutes depending on the size. This means a saving in fuel consumption over natural circulation especially as compared to the Velox or Steamotive types. The speed of starting make these forced circulation boilers more flexible and better suited for intermittent operations. Furthermore in the case of the recirculation type boilers, when the unit is banked one of the circulating pumps may be operated at low speed to maintain equal temperatures throughout. As in starting, less time is required when shutting down, and since there is less water in the unit, time is saved in draining. The "once through" types, of course, will not normally require any draining.

In case of tube rupture immediate shutdown is not necessary nor likely in forced circulation. This is a far more serious matter in natural circulation where a tube rupture requires immediate shutdown of the whole boiler. In the case of one large high pressure controlled circulation boiler installed at the Koppers synthetic rubber plant in Kobuta, Pennsylvania, it was found possible to operate for over three hours with water issuing from a ruptured tube. Since synthetic rubber is essentially a continuous process, the continued operation of this boiler prevented heavy loss in time and raw material which would have

otherwise occurred if this had been a natural circulation boiler. When shutting down due to a casualty, such as above, not only can the boiler be cooled down in less than half the time required for a natural circulation boiler, but the repair of a small tube requires very much less time than that needed for welding in a section of three or four inch tube. Thus the total outage is reduced.

4. Example Comparison of Forced and Natural Circulation Boilers

The only sizable forced circulation boiler installed in the United States during the 1940s for industrial plant processes was the installation at the Koppers Company Butadiene-Styrene Chemical Plant, at Kobuta, Pennsylvania. It will be recalled that the first installation at the Montaup Electric Company was for power purposes. These two constitute the only boilers using this principle to be placed in operation in this country until the present time.

The steam generating facilities at Kobuta, consisting of three natural - circulation and one forced-circulation boilers were built at a time when the demand for steel was at its peak. By government direction one of the four units was required to be forced circulation to economize on critical material. The plant began operation with one natural circulation unit in June 1943. The forced circulation unit, a Combustion Engineering Lamont type boiler, was placed in service in August of that same year. This entire plant was built in a minimum period of time and war regulations prohibited any but the most urgent alterations and refinements.

All four boilers were designed for 350,000 pounds per hour capacity at superheater outlet pressure of 725 psi and 750°F. A smaller boiler was later added. The boiler house was designed to house the natural circulation units and was some one hundred thirty feet high. As a result no attempt was made to cut down the size of the forced circulation unit and take advantage of one of its inherent advantages. The steam from the boilers goes to a 3500-KW turbo-generator which exhausts at 165 psi for chemical-plant process requirements. Total plant electrical load

is 14000 kw, the excess power feeding into the local utility. In contrast to central-power-station conditions where makeup water rarely exceeds 2 percent, and where even this is supplied by evaporation, boiler make-up at Kobuta has been as high as 60 or 70 percent because of the large process steam requirements.

Table 10 is a comparison of many of the characteristics of these two types of boilers:

TABLE 10 BOILER CHARACTERISTICS

	Forced Circ.	Natural circ.
Furnace volume, cu. ft.	25,400	29,900
Boiler heating surface, sq. ft.	4,510	20,640
Waterwall surface, sq. ft.	8,470	6,367
Superheater surface, sq. ft.	4,430	4,030
Total steam generating surface, sq. ft.	17,410	31,007
Economizer, sq. ft.	9,830	-
Air preheater surface, sq. ft.	38,900	38,600
Operating water volume, lb.	36,643	150,680
Steam drum	54" diam. x 28'-9½" long	60" diam x 30' long
Mud drum	-	42" diam x 29' long
Upper wall tubes	1¼"O.D. x 0.120" wall thickness	-
Lower wall tubes	1½"O.D. x 0.150" wall thickness	-
Front wall tubes		3¼"O.D. x 0.203" wall thickness
Rear and sidewall tubes		2½"O.D. x 0.170" wall thickness
Roof tubes		4 "O.D. x 0.213" wall thickness
Generating tubes		2½"O.D. x 0.148" wall thickness

During the latter part of 1943, all of 1944, and most of 1945, the plant was required to produce 1,300,000 pounds of steam per hour, with an installed capacity of 1,400,000 pounds per hour. This meant that outages had to be kept to an absolute minimum. During this entire period the forced circulation boiler was taken off the line every sixty days for inspection and test as compared with once every six months for the natural circulation units. This more frequent outage of the forced circulation boiler was necessitated by the water conditions and the fact that it was a new development. The Ohio River water is subject to wide variations in hardness and solids and during this two year peak load period the proper water treatment to prevent undue scaling was not discovered. The natural circulation boilers with four and one-quarter times as much water could more easily handle the solid concentration although two tube failures were attributed to this cause. The percent availability of the forced circulation unit during this War period was 95.08 percent as compared with 98.36 for the natural circulation units. Following the War this plant operated at a reduced capacity for several years, during which time the feed water problem was licked. Once again this plant is operating at high load and it is understood that the forced circulation boiler is operating at an even higher availability than the other units because it can be cooled down, acid cleaned, and put back on the line in two days as compared with three days minimum for the natural circulation units. The cost of cleaning is much less since only one quarter as much acid, water, and neutralizer is needed, to say nothing of the saving in time and labor of filling or draining.

Contrary to many predictions, the circulating pump has never been the cause of circulation failure in operations to date. This tends to bear out foreign claims and also experience with the Montaup boiler.

The forced circulation boiler has proved to be much more stable than the natural circulation from the standpoint of drum water level under varying loads. This is due to the constant rate of pumping, the circulation ratio being $5\frac{1}{2}$ to 1. In natural circulation a sudden increase in load causing a slight decrease in drum pressure causes an increase in volume of the steam bubbles which in turn causes a rise in water level in the drum. When not properly controlled this can result in carryover.

During the wartime operation the forced circulation boiler tended to slag up more quickly. Recent operation indicates little or no difference. This is attributed to the poor grade fuels available during the wartime operation and to the physical design of the waterwalls in which the designers left too small a space between tubes.

The efficiency of both the natural and forced circulation units is often give as 85 percent. When so calculated the work of the circulating pump in the forced circulation unit is taken as a total loss. This is not correct inasmuch as the largest part of this energy is returned to the water in the pump as heat. When properly calculated, then, the efficiency of this forced circulation boiler is a little higher than the others.

The natural circulation units are not equipped with economizers, but the forced circulation unit has one. This is logical since it helped improve the operation of this boiler and the increase in size

still did not exceed the space available determined by the natural circulation units.

The natural circulation boilers failed to produce steam at the predicted temperature. The temperature of the forced circulation unit has been kept down to prevent different steam temperatures in the turbine leads.

The biggest advantage of the forced circulation boiler in this installation, where constancy of steam supply is essential, is its previously mentioned ability to continue operation for a time after tube failure.

This particular forced circulation boiler has proved successful, all the difficulties of the wartime operation being subsequently corrected. Some features, however, suffer because of the restrictions and regulations imposed on all wartime construction. The design was a rush job and it is quite remarkable that it worked so well, especially at the high continuous loads prevalent during the first two years of operation.

IV. MARINE ADAPTATIONS OF FORCED CIRCULATION

1. General

Weight and space considerations are of far greater importance in marine boiler design than ashore. Ruggedness, ability to withstand sudden changes of load, and ease of starting are other prominent desirable features. Forced circulation would seem to lend itself quite readily to these conditions. Prior to World War II nearly all of the foreign merchant marines and navies were utilizing one type or another to a limited extent. In general the operation of these boilers was successful. During this period the most ardent advocate of forced circulation was the German Navy. Early problems in control were pretty well overcome, but tube failures were quite common in all their boilers, both forced and natural circulation. The War put an end to the development and it has only recently been taken up again to any degree.

In this country the U. S. Navy commenced experiments with forced circulation soon after World War II. Results of these experiments are mostly classified, but released information indicate gratifying results.

The obvious advantages of forced circulation for marine use, especially recirculation which does not rely on sensitive automatic controls, makes it only a matter of time until the few difficulties remaining are removed and its use becomes general.

2. S.S. "UCKERMARK"

Messrs. Blohm and Voss of Hamburg, Germany, built the first Benson boiler to be used on shipboard in 1929. It was installed a year later on the Hamburg-American Line ship "UCKERMARK". As originally built it was designed for 3200 psi operation according to the old design in which

the transformation zone lay flame-exposed. There were eleven radiant boiler coil tubes. On the maiden voyage to Canada several tube failures occurred. As a result the transformation zone was moved from the radiant to the convection part of the boiler. The operation during the next six years was highly successful.

In the meantime Benson boilers had been reduced in pressure with a resulting improvement in maintenance and operation. As a result this ship was reboilered in 1937 with two new Benson boilers of 710 psi and 788°F instead of the one critical pressure Benson boiler and three "Scotch" boilers.

Other marine Benson boiler orders were placed in 1938 for two boilers each for the liners "PRETORIA" and "WINDHUK". These were of 62,700 pounds per hour capacity operating at 1136 psi and 842°F. A number of these same type were later built for the German Navy.

3. S. S. "POTSDAM"

This liner was built for the North German Lloyd Company for their "Far East" service and entered service in July 1935. It was a twenty-one knot ship, six hundred fifty feet long and of 18,500 tons gross.

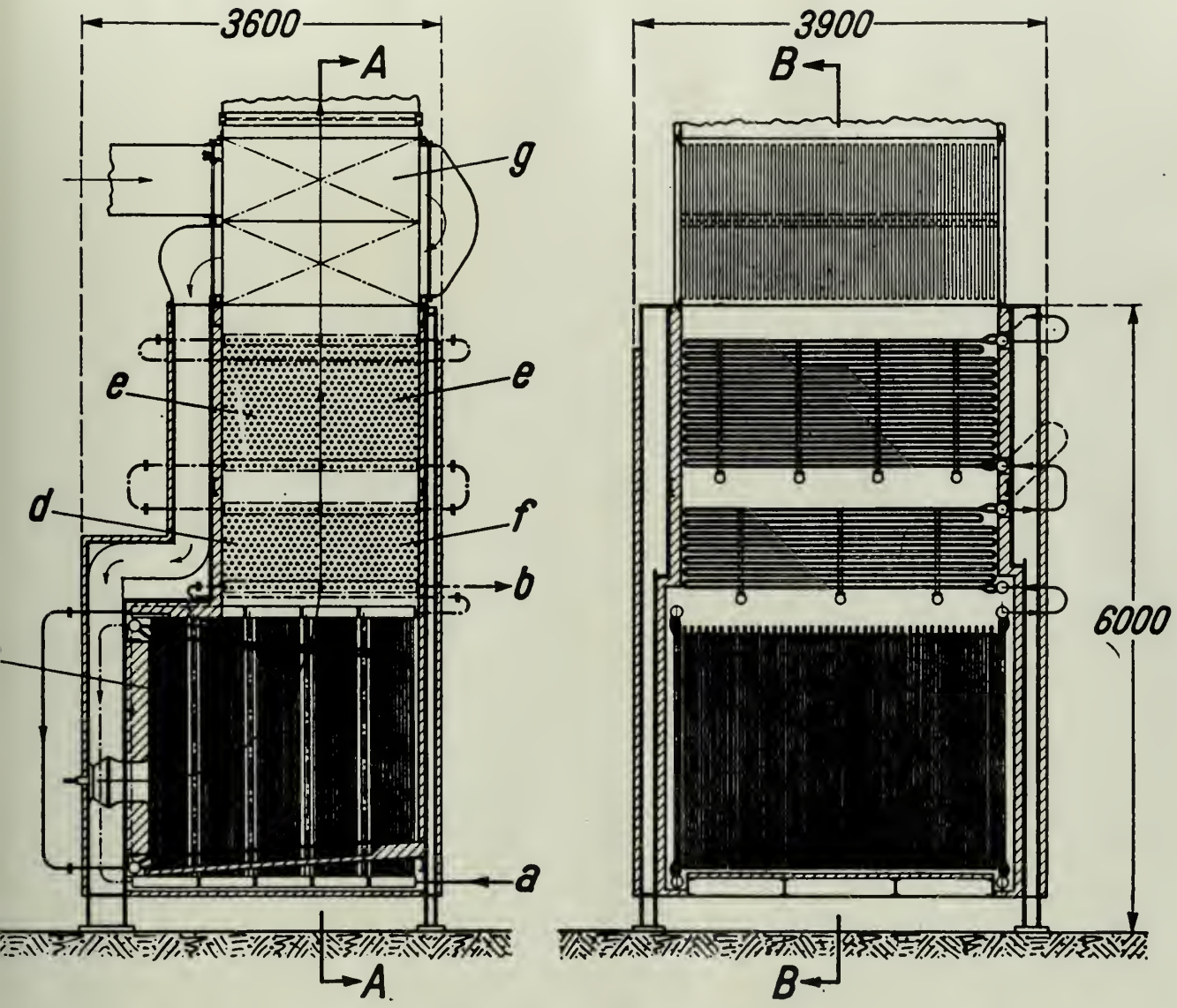
The boiler plant consisted of four Benson boilers, each with a total heating surface of 590 square meters or 6,300 square feet. These boilers furnished steam for high pressure turbines which in turn provided power for the electric drives. Much of the data upon which the design of this plant was based came from four years of experience with the Benson boiler on the SS "UCKERMARK".

On the "POTSDAM" each boiler contained twelve tubes in the radiation part of the boiler and twenty-two parallel tubes in other parts. Fifty three thousand pounds of steam per hour was generated at a normal pressure of 1200 psi and 880°F.

During trials, all the various maneuvers were carried out without any difficulty being experienced, the controls proving to be completely reliable. In operation the small evaporation surface prevented the water from foaming in case of suddenly increased output. The heat storage in the tubes, on the other hand, proved ample to meet large sudden demands for steam. The superheat temperature was much more constant under load changes than with conventional boilers. Salt deposition was no problem. The cleaning was normally done in port, at intervals of twelve to fourteen days. High inflows of salt were purposely tried during tests and it was found to cause no disturbance, provided they only lasted a short time. Any undue accumulation on the tubes could be easily washed off while at sea if found necessary.

Simple evaporation of sea water alone was not sufficient in this plant. As a result double evaporation was necessary which necessitated a sizable increase in the evaporation plant. Because of the high temperature and pressure, a closed circulation system was used to minimize the effect of free oxygen. The condensate was heated to boiling outside the condenser to eliminate free oxygen which was then treated chemically to prevent it from corroding the boiler tubes.

On the maiden voyage some tubes burst due to leakage into the condenser which could happen to any high pressure boiler. During the next few years these boilers were relatively free from tube failures and no trouble was ever experienced with the automatic controls.



. 18 —Sections through boiler of the Hamburg American ıer "Potsdam," 53,000 lb of steam per hour at 1200 lb pressure and 880 F.

feedwater inlet,
superheated steam outlet,
radiant heating surface,
(*d*) contact heating surface,
(*e*) salt deposit zone,
(*f*) superheater,
(*g*) air preheater.

4. S.S. "KUROSHIA-MARU"

Early in the 1930s the Japanese Kawasaki Shipyard in Kobe, purchased two German LaMont Steam Generators for shipboard use. These were highly successful and forced recirculation became relatively common in the Japanese Merchant Fleet and Navy.

Shortly before World War II the "Kuroshia-Maru", a large oil tanker, was built by the Kawasaki Yard. It was equipped with three Kawasaki IaMont marine, water tube, steam generators which were similar to the earlier German boilers, but with Japanese modifications and innovations. Each had a normal output of 37,500 pounds per hour and a maximum of 48,500 pounds per hour with a pressure of 315 psi at 662°F. The heating surface, minus the air preheater, was 3,529 square feet. The efficiency of one of these boilers reached 87.29 per cent on test ashore before installation.

The interesting feature in Japanese marine boiler operation concerns water contamination. One of the early German type forced circulation boilers, essentially similar to those above, was subjected to a series of tests in which the water contamination was deliberately increased to as much as 1165.5 grains per gallon. The boiler was then operated at overload capacity. No measurable carry-over resulted, though the testing equipment used could detect any amount above 0.58 grains per gallon.

From this series of tests the Japanese concluded that the steam marine plant could operate with ten times greater water contamination than is found in usual service, and in the case of small water tube boilers it is closer to 25 - 30 times. As a result from the middle 1930s on Japanese Naval and merchant ships were equipped with types of boilers in which the permissible salt content was much higher than our standards permitted.

As a comparison the U. S. Navy recommends maximum salinities as follows: Fire-tube boilers, 100 grains per gallon; water-tube boilers (two inch tubes), 50 grains per gallon; water tube boilers (less than two inch tubes), 30 grains per gallon.

5. S.S. "KERTOSONO"

This cargo ship belonging to the Rotterdam Lloyd Line was the first vessel to utilize the Sulzer marine monotube steam generator. The ship itself was built in 1922 by De Schelde at Flushing, Holland and was of 16,500 tons gross with a length of 445 feet.

Early in 1935 it was decided to convert this ship to high pressure operation. An increase in speed was desired. Since this ship maintained a tight schedule, speed in conversion was essential. The Sulzer boiler was decided upon and the rest of 1935 was taken up with building and testing this unit.

On February 16, 1936, the "Kertosono" entered the De Schelde yards for conversion. It was found sufficient to remove only one of the five six-ton capacity "Scotch" boilers ince the space thus left free was nearly three times as much as was needed for the Sulzer unit. This oil-fired boiler generated twenty-one tons of steam per hour - three and one-half times the capacity in one-third the space!

The new installation was designed to operate at 850 psi and 707°F. The total output was raised from 4500 to 5800 SHP and the speed increased from 13 to 15 knots. The plant was arranged so that it was possible in an emergency to work directly with the low pressure steam from the remaining four boilers. This included the new primary turbine installed

simultaneously with the monotube boiler. On the existing turbines only minor changes to the blading was found necessary. Among the automatic controls installed for the operation of this boiler was a regulating gear which permitted the regulation of the monotube boiler and also the turbine by means of a single hand wheel. On subsequent trials this greatly facilitated maneuvering. "Full speed ahead" to "full speed astern" was effected in fifteen - twenty seconds.

The guarantees given for the steam generator were easily fulfilled, and efficiencies up to 94 percent were measured. Between half and full load and including the auxiliaries, an efficiency of over 90 percent was obtained.

The entire conversion was completed by March 6, 1936, just under three weeks. In later operation this plant proved successful and reliable.

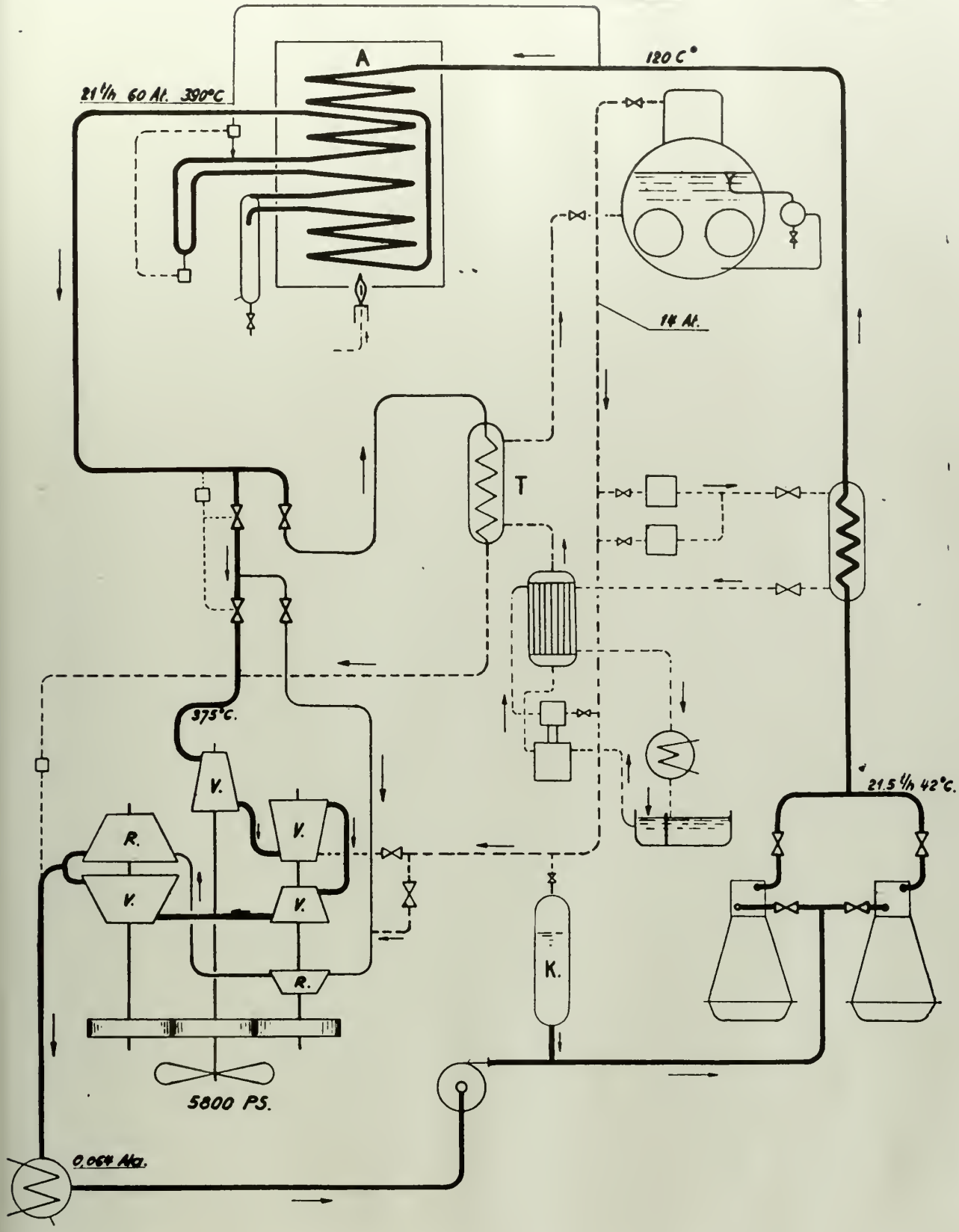


Fig. 19 —Circuit of Sulzer Boiler Plant in S.S. "*Kertosono.*"

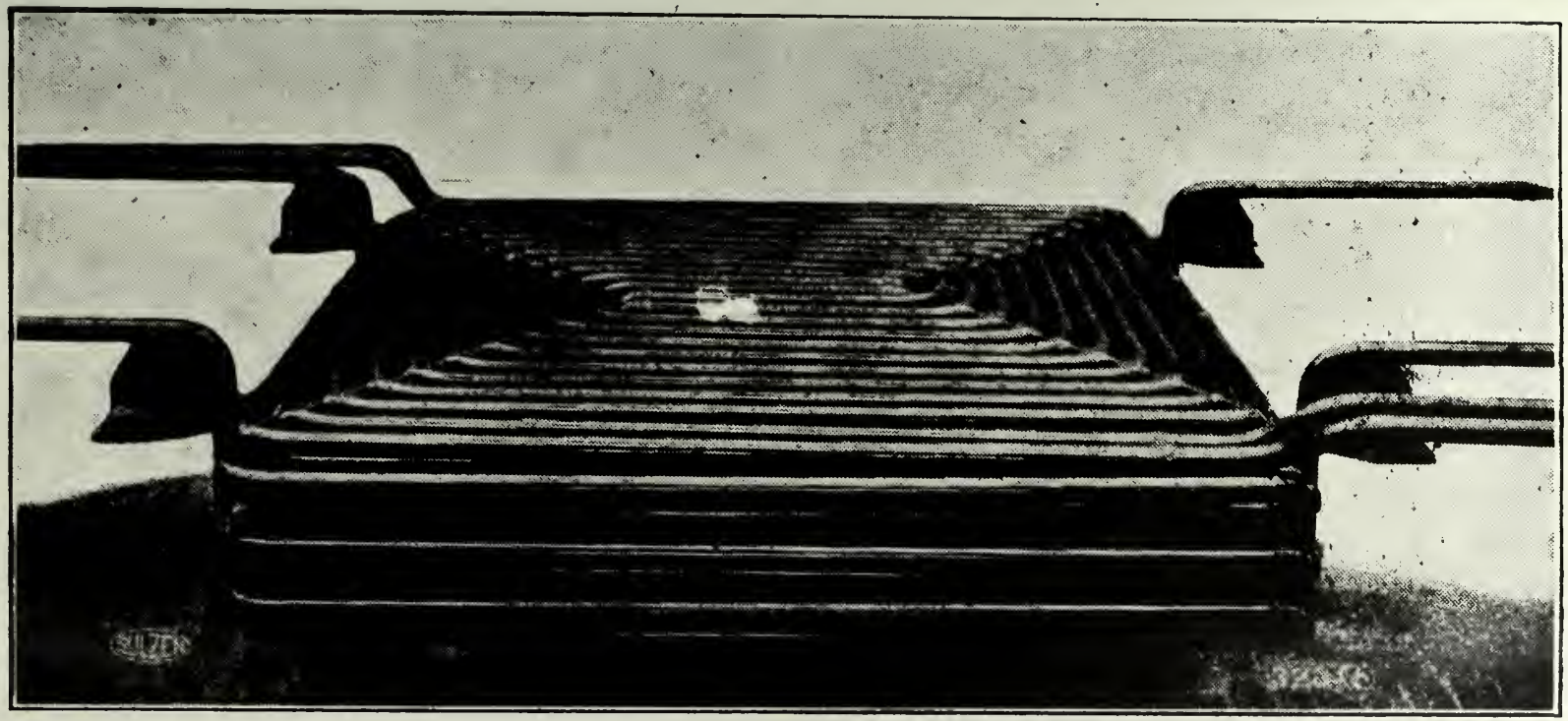


Fig. 20 —S.S. "*Kertosono*" Boiler Coil.

6. S.S. "ATHOS II"

The Velox boiler is well suited to naval use because in addition to its light weight and small space requirements, it has an almost instantaneous response to sudden load fluctuations and a very rapid starting. Tests on one of these boilers showed that it was possible to bring the boiler up to full operating pressure from cold in six minutes eleven seconds and that in a further one minute twenty seconds full load evaporation was reached.

A number of these units have been installed aboard ship, mostly naval vessels, so that information is difficult to obtain. However, the first passenger ship to be so equipped was the French liner "Athos II" owned by Cie. des Messageries Maritimes. This ship was of 21,000 tons gross. Originally this ship, operated by a twin screw, had seven "Scotch" boilers working at 237 psi and 570°F., along with two steam turbines of 5,000 H.P. One "Scotch" boiler was removed and replaced by a Velox generator operating at 725 psi. and 840°F. with a capacity of 77,175 pounds per hour. A high pressure turbine was also added, exhausting to the existing steam pipe range, supplying the two 5000 h.p. turbines. In addition a new L.p. condensing turbine was added in parallel to utilize the greater amount of steam present. With no increase in boiler and power plant space, the total engine power was raised from 10,000 to about 20,000 h.p. and the speed upped from 15-16 to 20 knots.

to 24 hours.

7. S.S. "NICEA" and Other LaMont Installations.

The use of LaMont waterwalls to increase conventional boiler capacity was discussed in Chapter II. These waterwalls have also been installed for the same purpose in a number of ships where suitable. Probably the most publicized cases were those installed in the Hamburg-American liners "Albert Ballin" and "Deutschland". The auxiliary boilers in these ships were found to be of insufficient capacity for the supply of steam to the turbo generators, winches, refrigeration, etc. while the steamer was in port. As a result these waterwalls, which were installed in 1931, brought the uptake temperatures due to overloading down from 930°F to about 570°.

A complete LaMont installation consisting of two powdered-coal fired boilers was installed in the "Nicea". From the accompanying figure it can be seen that these boilers were fired by inverted flame type burners and contained a radiant evaporating section followed by a coil shaped convection area. These were not large boilers and their rate of evaporation was low. Pertinent data:-

Steam pressure	255 psi
Steam temperature	626°F
Evaporating Surface	710 sq. ft.
Superheating Surface	312 sq. ft.
Steam output per boiler	11,000 lbs. per hr.
Average rate of evaporation	15.5 lbs/sq ft/ hr.

A large marine LaMont plant gives the following data by comparison:

Steam pressure	750 psi
Steam temperature	806°F
Evaporating surface	2,320 sq. ft.
Superheating surface	1,830 sq. ft.
Steam output	88,000 lbs. per hr.
Average rate of evaporation	38 lbs/sq. ft./hr.

The total service weight of this boiler including water is 52,800 pounds which is about 0.6 pound per pound of steam generated per hour. In spite of this small weight the efficiency of this unit tested at 86.2%.

LaMont type waste heat boilers have found extensive use in conjunction with marine diesels. They have proved quite satisfactory in this service. Early installations occurred about 1930 on the motorships "Toulouse", "Modjokerto", and the motor tanker "Svithiod". Their use extended rapidly after this with the increase in use of diesels afloat.

Advanced boiler design was characteristic of the late German Navy vessels. Steam pressures of over 1000 psi and temperatures of 700° to 850°F were the rule rather than the exception. LaMont, natural-circulation, and Benson types were all in use, the latter type on small destroyers and coastal craft. The LaMont installation best known in this country was that on the large heavy cruiser "Prinz Eugen". This installation was highly successful according to available reports as were the installations on the smaller ships.

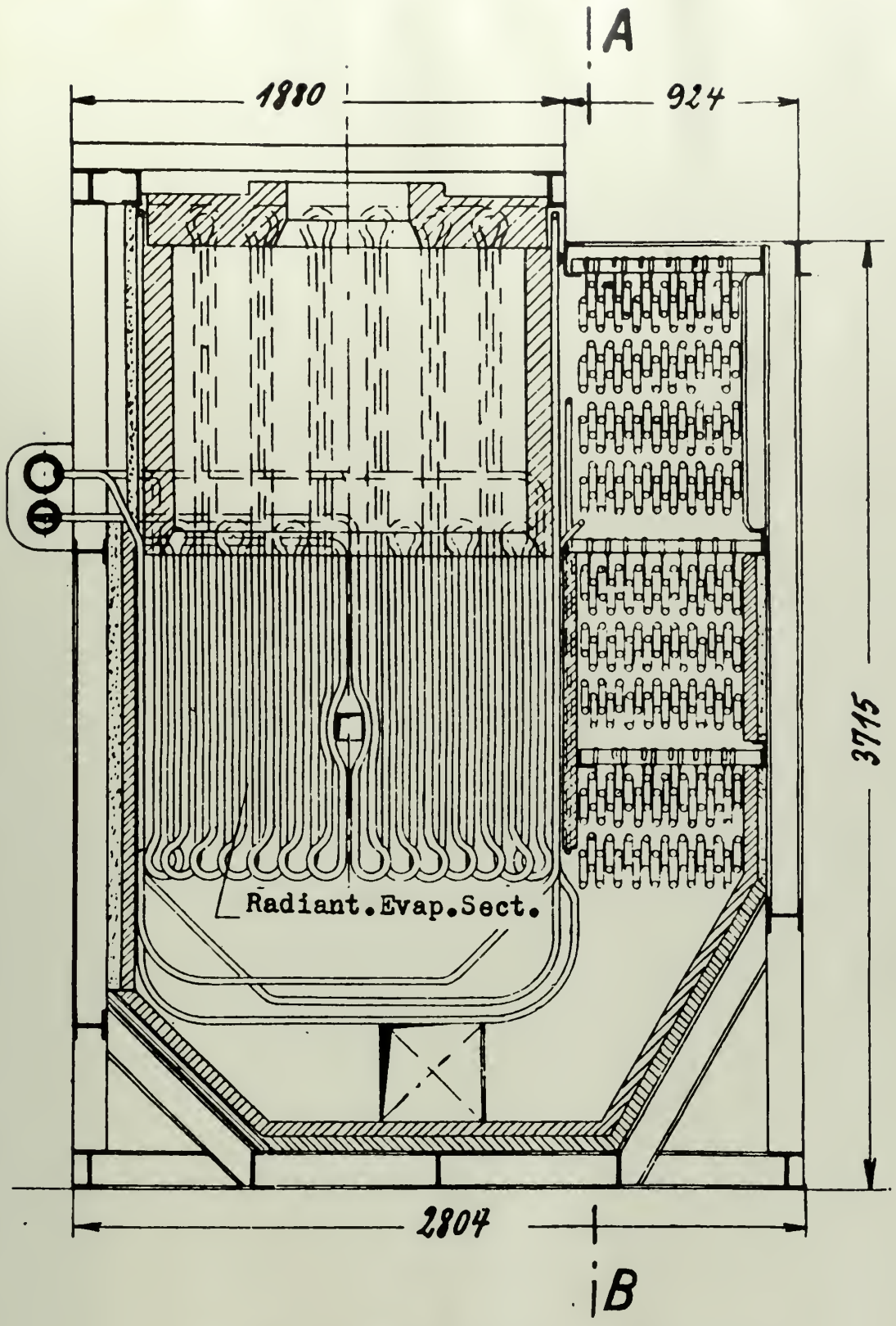


Fig. 21 —La Mont Boiler in S.S. "*Nicea.*"

In 1938 the British Navy built the destroyer H.M.S. "Ilex" which was powered by a LaMont type boiler plant. This was a successful installation, especially the quick starting feature. As a result of the experience with this ship, five corvettes were powered with these boilers shortly after the outbreak of World War II. These installations were disappointing, the particular design used being an adaptation and unsuited for the type of service required on these ships. The war brought a halt to other projected trial installations, and while it is known that the Royal Navy has again taken up forced circulation, security restrictions have prevented the publishing of any definite information on the subject.

Both the Japanese and the French navies had done considerable work on the LaMont type boilers by the outbreak of the War. It is likely that the French have again started work on this development, but no current information is available.

8. S.S. "CONTE ROSSO"

The "Conte Rosso" was a luxury passenger ship of 18,500 gross tons built in 1922 for Messrs. Lloyd Triestino of Trieste. In March 1936, this vessel was in the hands of Messrs. Cantieri Reuniti dell Adriatico for general overhaul and redecoration.

As part of the reconstruction work it was arranged to increase the power developed by 5,000 s.h.p. from 17,000 s.h.p. to 22,000 s.h.p. and by this means to increase the speed of the ship by one and one-half knots. This was to be done by the installation of the first Loeffler boiler to be installed on board ship.

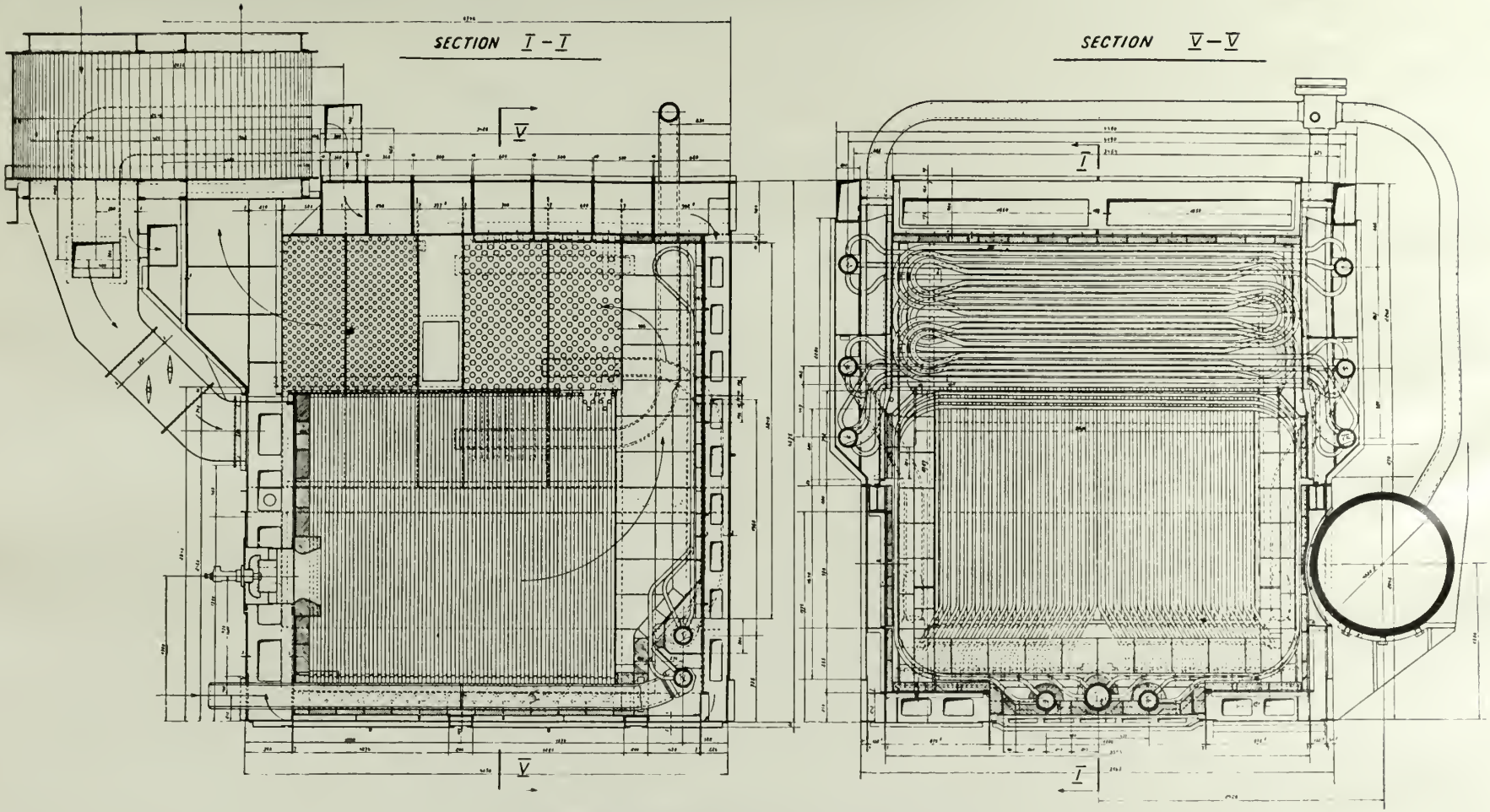


Fig. 22—Loeffler Boiler in S.S. "*Conte Rosso*."

The original boiler plant included six double and two single "Scotch" boilers generating steam at a pressure of 200 psi and 572°F. One single ended 16,000 pounds per hour boiler was removed and the Loeffler boiler installed in its place. The normal capacity of this boiler was 44,000 pounds of steam per hour at a pressure of 1,850 psi and 890°F. The maximum rate provided was 55,000 pounds steam per hour. Two new high pressure turbines were installed, one for each shaft. They worked in series, the first exhausting at 660 psi to the second where the exhaust steam was reheated and delivered to the existing steam supply main at 200 psi and 572°F.

The total weight of this boiler with all its appendages was only 53 tons and with an output of 22 tons of steam per hour this represented a weight of boiler per ton of steam produced per hour of 2.6 pounds. This was very good since the efficiency was 90%. A total of 28.55% of the heat released in the furnace, appeared as s.h.p. This gives less than 9000 BTUs per s.h.p. or under 0.51 pounds of fuel per s.h.p.

9. Sural Boiler (Pressure Fired)

Three views are presented of the Sural boiler developed just prior to the War for the French Navy. It is small, compact, light in weight and has a high capacity. In order to achieve all these features the combustion rate had to be increased and also the rate of heat transfer. This was done by the employment of a high pressure combustion chamber which circulated the gases at high velocity and made possible a steam output of 156,800 pounds per hour with less than 2,153 square feet of heating surface. To protect the tubes, vigorous water circulation was

essential. This was usually done by a pump although natural circulation was also tried with fair results. If forced circulation is used a separator must be provided to separate the steam and water.

The most novel feature is the use of flue gas, drawn directly from the boiler, to drive a turbine coupled to a blower which provides the compressed air for combustion. In this respect it is similar to the Velox design. The high combustion pressure allows a reduction in combustion chamber size and provides satisfactory operating conditions for the gas-turbine driven blower. No air heater is needed since the compression of the combustion air raises its temperature to a high value. A number of these units have been built, the early ones with centrifugal blowers, the later ones with axial-flow types. The pressure of the air delivered by the turbo-blower units can be varied from 14 to 36 psia.; the normal pressure in the combustion chamber being of the order of 21 psia at full load. The temperature of the gas turbine outlet runs about 460°F.

Figure 23 is a diagrammatic layout of an early type Sural boiler of 156,800 pounds per hour capacity. The total weight is only 51 tons including turbo-compressor, separator, and circulating pump. Figures 24 and 25 are views of a later boiler of this type with the same capacity in which the design has been simplified to reduce cost and facilitate boiler cleaning.

These boilers are included because they not only illustrate a type of forced circulation but in addition present the feature of true pressure firing which has additional possibilities with respect to reduction of size and increase of output.

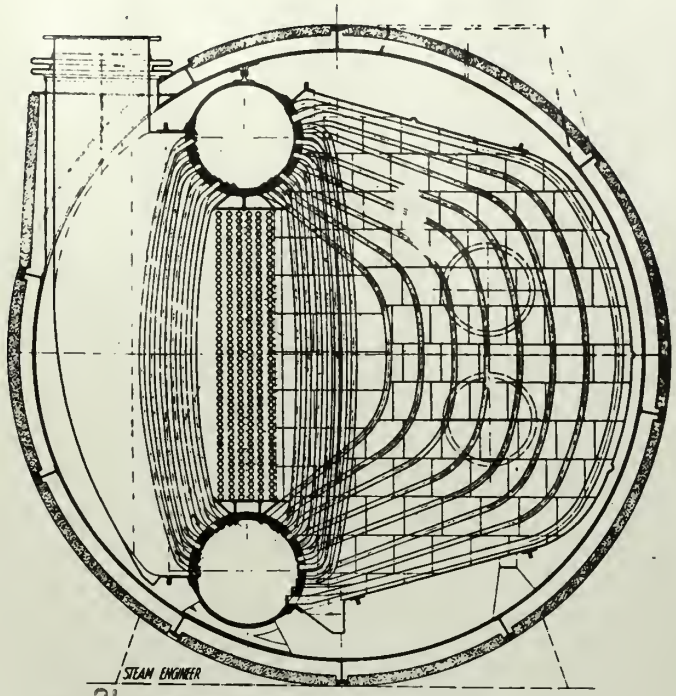


FIG. 24—*Sectional elevation of latest type of Sural boiler.*

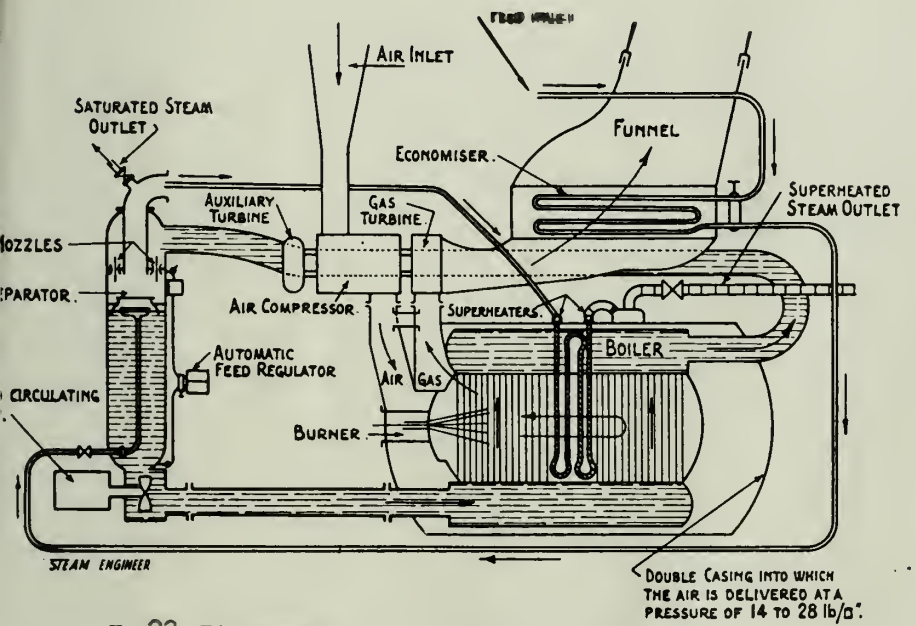


FIG. 23—*Diagrammatic lay-out of early type of Sural boiler.*

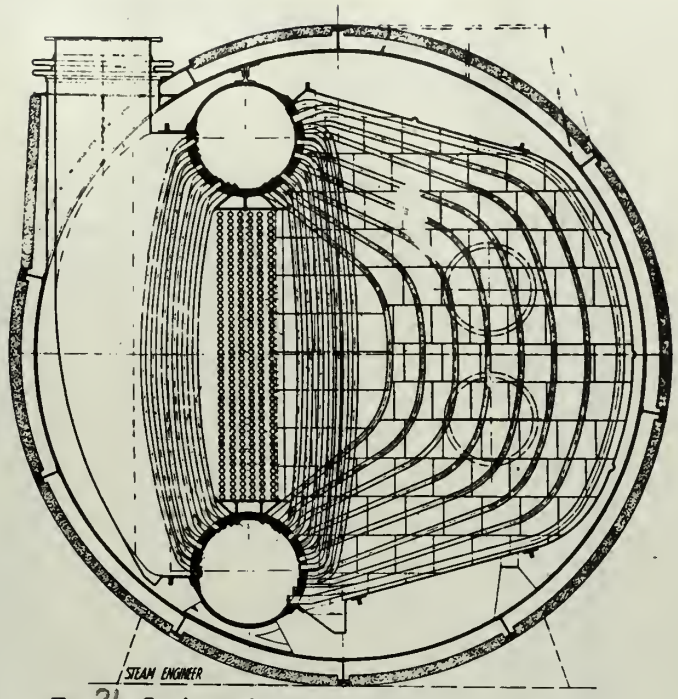


FIG. 24—*Sectional elevation of latest type of Sural boiler.*

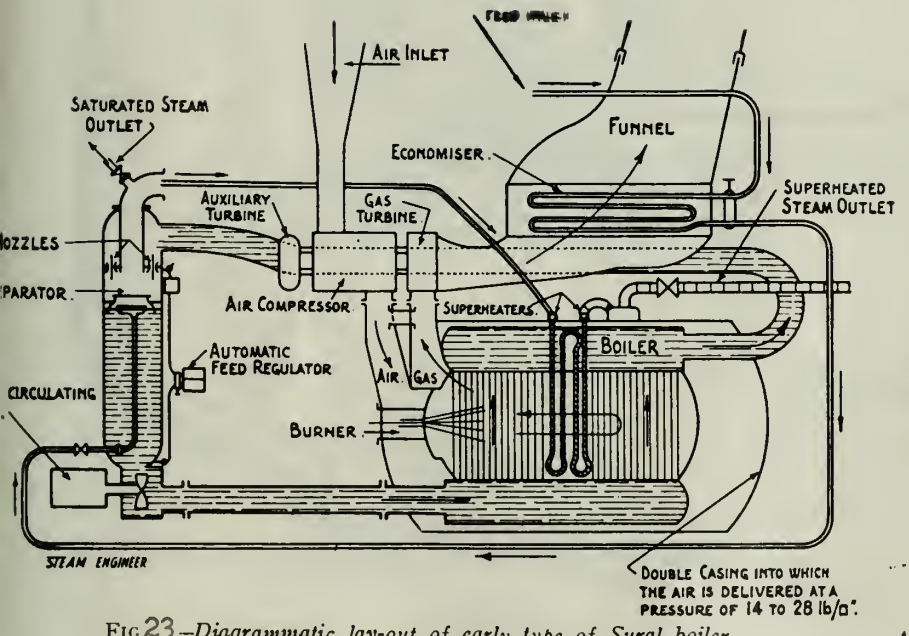


FIG. 23—*Diagrammatic lay-out of early type of Sural boiler.*

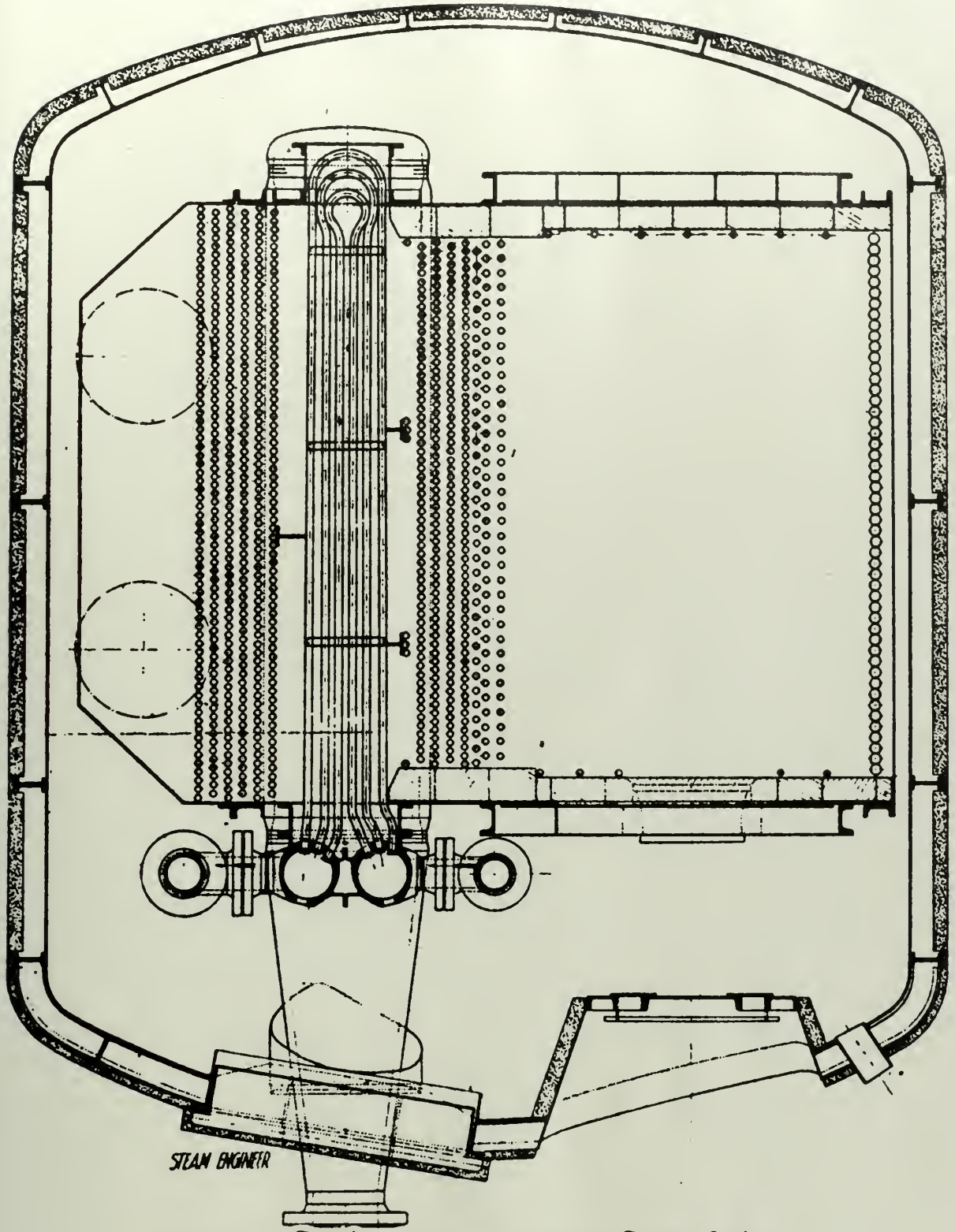


FIG25 —*Sectional plan of new Sural boiler.*

10. Schmidt-Hartmann Marine Boiler

Figure 26 shows a Schmidt-Hartmann ship's boiler installed prior to the War. The operating data is as follows:

	Normal	Maximum
Steam output, pounds per hour	13,970	18,150
Operating pressure, psi	738.4	738.4
Primary pressure, psi	1250	1462
Final steam temperature °F.	907	932
Boiler overall eff. percent	87.0	83.0

The steam was supplied to a quadruple expansion engine of 1800 I.h.p. The figure shows a division wall in the center of this boiler. This permitted more efficient operation while in port and improved reliability.

All tubes of the primary circuit were made of carbon steel. The superheater tubes and the drums were made of low alloy copper-molybdenum steel, which is comparatively light in weight.

The evaporator tubes are slanted part one way and part the other. This prevents emergence of too much evaporator surface due to pitching of the ship.

These boilers have relatively large thermal and water storage capacities and their operation is not much affected by load fluctuations due to maneuvering.

10. Calculation Sheet

Figure 20 shows a Total Pressure Calculation Sheet for a Forced Draft Boiler. The operating data is as follows:

	Normal	Maximum
Steam output, pounds per hour	14,000	20,000
Superheat pressure, psi	250.4	250.4
Primary pressure, psi	1250	1400
Final steam temperature °F.	700	750
Boiler overall efficiency	81.0	80.0

The steam was regulated to a quantity equation weight of 1800 l.b.s. The figure shows a circular wall in the center of this boiler. This pattern have efficient operation which is good and improved reliability.

All tubes of this primary circuit were made of carbon steel. The superheater tubes and the drums were made of low alloy carbon-molybdenum steel, which is comparatively light in weight.

The evaporator tubes are placed part one way and part the other. This provide maximum of the more evaporative surface due to pinching of the gas.

These boilers were relatively large thermal and water storage capacities and their operation is not much affected by load fluctuations due to superheating.

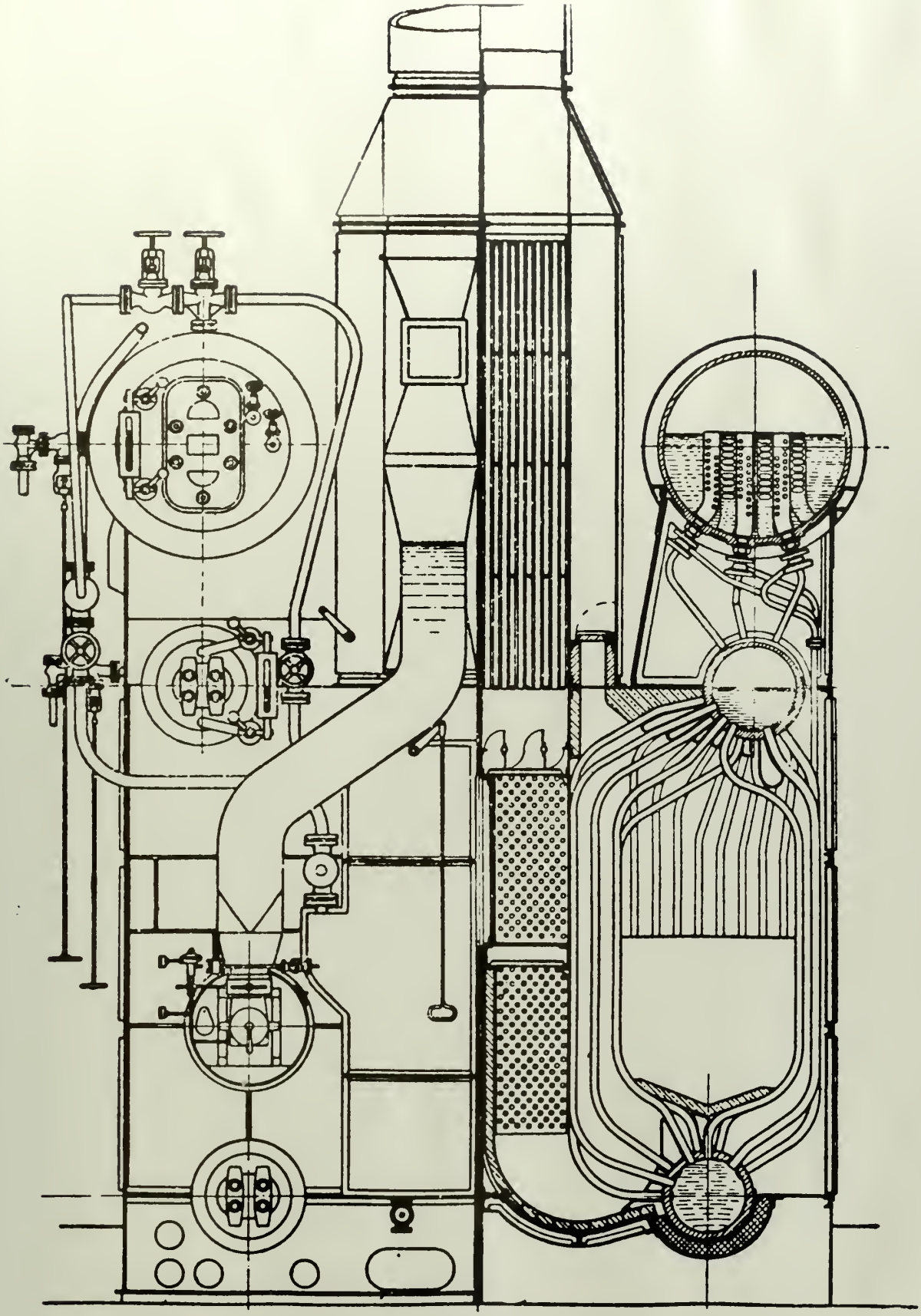


Fig. 26 —Schmidt-Hartmann Marine Boiler.

11. Forced Circulation in the U. S. Navy

Shortly after the last War the U. S. Navy sponsored an extensive series of experiments on high pressure forced circulation boilers for shipboard use. These were of both the Steamotive and LaMont types. Most of the data on these tests is of a classified nature, but in recent months some of the information has been made available. Only released information will be discussed here.

As a result of these experiments, modified LaMont type boilers, built by the Combustion Engineering Superheater Company, are currently being supplied for naval use. These units are designed to generate at full power approximately 145,000 pounds of steam per hour at 1200 psig and 950°F. Table 11 gives pertinent performance and physical data for these boilers as compared with a natural circulation boiler designed for the same capacity. It will be noted that the box volume of one controlled circulation unit including the two circulating pumps is 523 cubic feet less than the comparable natural circulation boiler. This means a weight saving of 36,000 pounds per boiler or for this particular four boiler installation, 72 tons per ship.

TABLE 11

	Natural Circulation	Forced Circulation
Number of boilers per ship	4	4
Rating, per cent	100	100
Total Evaporation lbs./hr.	145,000	145,000
Equivalent Evaporation lbs./hr	186,300	186,300

	Natural Circulation	Forced Circulation
Superheater Outlet Pressure psig	1200	1200
Drum Pressure psig	1275	1275
Total Steam Temperature, Degrees F.	950	950
Feed Water Temperature, Degrees F.	246	246
Efficiency, percent	81.7	82.3
Firing Rate, lbs/hr	11,970	11,875
Excess Air, percent	15	15
Furnace Volume, cubic feet	518	305
Boiler Heating Surface, square feet	2420	2265
Water Wall Surface, square feet	230	270
Total Generating Surface, sq. ft.	2650	2535
Superheater Surface, square feet	990	770
Economizer Surface, square feet	3600	3630
Total Heating Surface, square feet	7240	6935
Heat Release per cu. ft. Furnance Volume Kb/hr	427	721
Heat Release per sq. ft. Total Heat Surface Kb/hr	30.6	31.7
Width	16' - 0"	14' - 0"
Height	18' - 0"	17' - 7"
Depth	11' - 0"	10' - 9"
Box Volume, cubic feet	3168	2645
Volume Decrease, cubic feet	-	523
Total Dry Weight per boiler, pounds	134,000	104,000
Total Wet Weight per boiler, pounds	146,000	110,000
Weight Saving, pounds (wet)	-	36,000
Weight Saving, percent (wet)	-	24.7

In Table 12 pertinent performance and design data are tabulated for a natural and a controlled forced circulation boiler designed for a full power evaporation of 275,000 pounds of steam per hour. For this particular installation a saving in the depth dimension was of considerable importance. It will be noted that the forced circulation unit has a depth of 11 feet 2 inches compared to 14 feet 4 inches for the natural circulation boiler. In this installation the saving could amount to 25 feet in length of the machinery spaces, and a weight saving of 41,500 pounds per boiler or 166 tons for the eight boilers in the ship.

TABLE 12

	Natural Circulation		Forced Circulation	
No of Boilers per Ship	8		8	
Condition	Cruising	Full Power	Cruising	Full Power
Rating percent	45.0	100	45.0	100
Act. Evap lbs/hr	122,000	271,000	122,000	271,000
Drum Press. psig	1218	1285	1218	1285
Superhtr Outlt Press psig	1200	1200	1200	1200
Tot. Steam Temp deg F	930	950	930	950
F.W. Temp deg. F.	245	245	245	245
Excess Air, percent		18.5		18.5
Efficiency, percent	87.9	83.7	88.0	83.9
Oil Firing Rate lbs/hr	8850	21,200	8850	21,200
Air Flow, CFM	34,300	82,300	34,300	82,300
Gas Flow, lbs/hr	155,000	371,000	155,000	371,000
Gas Temp leaving Econ. deg. F.	375	565	375	565

Release Rate per sq. ft. Tot. Heat Surface Kb/hr	11.6	27.8	13.2	31.5
Release Rates per cu. ft. Furn. Vol. Kb/hr	170	406	274	656

Boiler Heating Surface, sq. ft.	5780	3680
Water Wall Heating Surface, sq. ft.	405	440
Superhtr Heating Surface, sq. ft.	1440	1320
Economizer Heating Surface sq. ft.	6340	7015
Tot. Heating Surface, sq. ft	13,965	12,455
Furnace Volume, cu. ft.	964	597
Height to Econ. Outlet	21'-6"	21'-6"
Height to Drum Shell	20'-4½"	18'-6"
Width	18'-1"	18'-1"
Depth	14'-4"	11'-2"
Box Volume, cu. ft.	5560	4330
Weight Dry, pounds	190,600	154,200
Weight Water, Steaming, lbs	15,400	10,300
Weight Steaming, pounds	206,000	164,500

From the data in these two tables it is evident that a forced circulation LaMont type boiler for marine use will weigh about 75 percent as much as a comparable natural circulation marine boiler. This is generally the case for such boilers designed for 1000 psig and over with evaporations of over 100,000 pounds steam per hour. It is interesting to note that at 120 percent full power the forced circulation boiler in Table 12 had sodium chloride injected periodically into the

steam drum until the total solids in the boiler water reached 1604 ppm and a chlorinity of 586 ppm. The maximum test stipulations were 1150 ppm total solids of which 515 ppm must be chloride. Steam purity data under these conditions gave a steam carryover of 0.04 percent moisture or about one-sixth of the maximum allowable of 0.25 percent. The above test was performed at the Naval Boiler and Turbine Laboratory.

V. CONCLUSION

The forced circulation principle represents as big a step forward in boiler design and performance as the advent of the watertube boiler over the firetube. For the present and immediate future the recirculation type, especially the LaMont design, seems to hold the most promise. Eventually as the automatic controls become more accepted and a truly successful and convenient method of getting rid of salt deposition is devised, the "once through" type should take over the field of steam generation.

The United States has lagged behind the other nations in adopting forced circulation even though the basic principles were discovered here. This is partly due to the lack of a strong economic pressure such as experienced by continental Europe in the 1920s and partly due to the conservatism which our mass production methods engender. This country is now making vast strides to catch up. Since the War the U. S. Navy has had both Babcock and Wilcox and the Combustion Engineering - Superheater Company actively engaged in marine forced circulation development. The power companies in their thirst for greater and greater efficiency with its high temperatures and pressures are beginning to realize the savings in initial cost, maintenance, etc. which forced circulation offers. The Montaup and Kobuta installations operating over the past ten and nine years respectively have shown the reliability of the LaMont type boiler with its high efficiency and economies. Consequently, after the initial flurry of peacetime building following the War, these power companies commenced ordering forced circulation units of the Combustion Engineering - LaMont type. The first order was in August 1950. By

November 1951 there were eighteen on order ranging in size from 750,000 to 1,450,000 pounds of steam per hour, and in design pressures from 1670 to 2650 psig. All will employ reheat of 1000 or 1050°F and initial steam temperatures run from 1000 to 1100°F. As of April 21, 1952, the number of these units on order runs to thirty-six. Table 13 is a partial list of these units and their ratings.

TABLE 13

Power Company	Capacity Per Unit (lbs.per hr)	Design Pressure (psi)	Steam Temperature (° F)
Virginia Elec. & Power Co.	750,000	1670	1000 - 1000
Duke Power Company	900,000	2010	1000 - 1000
Wisconsin Elec. Power Co.	795,000	1775	1000 - 1000
Southern Cal. Edison Co.	920,000	2100	1000 - 1000
Cleveland Elec. Illum. Co.	875,000	2030	1000 - 1000
Public Service Elec. & Gas Co.	1,015,000	2650	1100 - 1050
Wisconsin Elec. Co.	795,000	2050	1050 - 1000
Philadelphia Electric Co.	1,450,000	2075	1000 - 1000
Consumers Power Co.	1,050,000	2300	1050 - 1000

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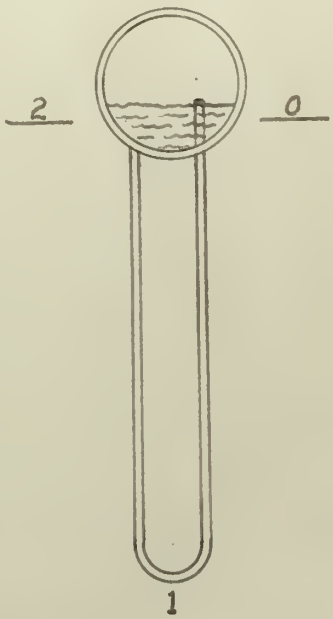
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APPENDIX "A"

The following derivation illustrates the relationship between the General Energy Equation and the equation used in boiler design work by the Combustion Engineering Company, Babcock and Wilcox Company, and with slight variations by the other boiler manufacturers.



Between points 0 and 1

$$\frac{g}{32.17}(z_o - z_o) + \frac{V_o^2}{64.35} + \frac{p_o}{d} + E_o + {}_oW_1 + {}_oq_1 = \frac{g}{32.17}(z_o - z_1) + \frac{U_1^2}{64.35} + \frac{p_1}{d} + E_1$$

There is no mechanical pump so the ${}_oW_1$ term goes out, and the $(E_1 - E_o)$ and ${}_oq_1$ terms may be represented by ${}_o(E_f)_1$ since any increase in internal energy is due to friction and turbulence and the heat loss ${}_oq_1$ is minor

$$\text{Therefore} \quad \frac{g}{32.17}(z_1 - z_o) + \frac{p_o - p_1}{d} = {}_o(E_f)_1 + \frac{U_1^2 - U_o^2}{64.35}$$



Between points 1 and 2

The same reasoning as applied above will give:-

$$\frac{g}{32.17}(Z_2 - Z_1) + \frac{p_1 - p_2}{d} = {}_1(E_f)_2 + \frac{U_2^2 - U_1^2}{64.35}$$

Between points o and 2

The two above equations may now be combined and simplified as follows:-

$$\frac{g}{32.17}(Z_1 - Z_o) + \frac{g}{32.17}(Z_2 - Z_1) + \frac{p_o - p_1}{d} + \frac{p_1 - p_2}{d} = {}_o(E_f)_2 + \frac{U_1^2 - U_o^2}{64.35} + \frac{U_2^2 - U_1^2}{64.35}$$

$Z_1 - Z_o = -(Z_2 - Z_1)$ $\therefore$ Z terms drop out

$p_o - p_1 = d_D \times Z$ pressure increasing.

d_D = density of downcomer.

$p_1 - p_2 = d_M \times Z$ pressure decreasing.

d_M = density of steam - water mixture.

$$\therefore \frac{(d_D - d_M)}{d} Z = {}_o(E_f)_2 + \frac{U_2^2 - U_o^2}{64.35} = R$$

$d = d_s$ since this calculation is referred to saturated water.

R = total resistances in the circuit. Since these are all due to velocities in the circuit, each resistance term may be written in terms of a velocity.

$$\therefore \quad \frac{(p_1 - p_2)}{\rho} = \frac{(v_1^2 - v_2^2)}{2} + g(z_1 - z_2)$$

The equation as used by the designer for the simple U-circuit shown in the sketch is:-

$$\therefore (\frac{d_D}{d_s} - \frac{d_M}{d_s})Z = \frac{1}{2} \times \frac{U_1^2}{2g} + \frac{U_2^2}{2g} + \frac{4fLU_D^2}{d \times 2g} + \frac{4fLU_M^2}{d \times 2g} + \frac{U_1}{g}(U_2 - U_1)$$

The left hand side represents the available head.

$\frac{1}{2} \times \frac{U_1^2}{2g}$ is the entrance loss. U_1 is the entrance velocity.

$\frac{U_2^2}{2g}$ is the exit loss. U_2 is velocity at exit.

$\frac{4fZU_D^2}{D \times 2g}$ is downcomer tube friction loss. $U_D = U_1$ if downcomer is all water.

$\frac{4fZU_M^2}{D \times 2g}$ is tube friction loss in riser. U_M is mean velocity of the steam - water mixture.

For mixtures each of the above resistance heads is multiplied by the relative density of the fluids.

$\frac{U_1}{g}(U_2 - U_1)$ is the acceleration loss.

All of the above velocities are in fps.

The equation as used by Lee Anderson for the Langen Cardioid system is in the
usual form:-

$$\therefore \frac{dv}{dt} - \frac{v^2}{r} = \frac{f}{m} = \frac{F}{m} \cdot \frac{r}{r} = \frac{GMm}{r^2} \cdot \frac{1}{m} \cdot \frac{r}{r} = \frac{GM}{r^2} \cdot \frac{r}{r} = \frac{GM}{r^3} \cdot r = \frac{4\pi^2}{T^2} \cdot r = \frac{4\pi^2}{T^2} \cdot (x_2 - x_1)$$

The left hand side represents the motion in orbit.

$\frac{v^2}{r}$ is the Acceleration Line. v is the velocity reference.

$\frac{v^2}{r}$ is the arch Line. v is velocity at orbit.

$\frac{4\pi^2 r}{T^2}$ is Centripetal Acc Circular Line. $\frac{v^2}{r} = \frac{4\pi^2 r}{T^2}$ if
diagonals is all above.

$\frac{4\pi^2 r}{T^2}$ is Time Position Line to Sun. $4\pi^2$ is mean velocity
of Lee Anderson - Wales Relation.

The distance scale of the above resistance means is multiplied by the
relative density of the Circle.

$\frac{1}{2}(v_2 - v_1)$ is the Acceleration Line.

All of the above relationships are for r_2.

APPENDIX "B"

For clean steel or cast iron pipes which as a rule fall in the rough pipe category, McAdams in his second edition of "Heat Transmission" (34) (page 120) has a formula which approximates the data of a number of observers for the value of friction factor f .

$$\frac{1}{\sqrt{f}} = 3.2 \log_{10} (R_e \times \sqrt{f}) + 1.2$$

Multiplying through by $\sqrt{f}$ and rearranging this becomes

$$\sqrt{f} \quad 3.2 \log_{10} (R_e \times \sqrt{f}) + 1.2 = 1$$

This is a trial and error equation. On the water side of the circuit for the 1.15 O.D. (0.95I.D.) inch tube at 1000 psi used in the calculation of mean temperatures and temperature gradients on page 85 the Reynold's number is:-

$$\frac{D_i U d}{u} = \frac{0.0793 \times 16000 \times 46.3}{0.235} = 250,600$$

Now substitute this into McAdam's equation above for various values of friction factor f. As indicated below the equation balances when f = .0046.

f = .0046

$$\sqrt{.0046} \quad 3.2 \log_{10} (250600 \times \sqrt{.0046}) + 1.2 = 1$$

$$.06783 \quad 3.2 \log_{10} (16998) + 1.2 = 1$$

$$1 = 1$$

On the 4.0 O.D. (3.48) I.D.) inch tube for the same pressure the Reynold's Number is:

$$\frac{D_1 U d}{u} = \frac{0.29 \times 22880 \times 46.3}{0.235} = 1{,}310{,}000$$

$f = .0036$

$$\sqrt{.0036}\ \ 3.2 \log_{10} (1{,}310{,}000 \times \sqrt{.0036}) \neq 1.2 = 1$$

$$.06\ \ 3.2 \log_{10} (78{,}600) \neq 1.2 = 1$$

$$1.01 = 1 \quad \text{(approximately)}$$

The correct Reynold's Numbers for wet steam cannot be determined with any degree of accuracy because of the lack of data on the viscosities of such a mixture.

Although the above examples indicate values of f much lower than the assumed $f = .006$, it should be remembered that the calculations are for clean pipes. This condition is seldom the case in boiler tubes since scaling and corrosion cause greater roughness and often an appreciable decrease in inside tube diameter. This, in turn, adds considerably to the friction factor. For badly corroded or tuberculated pipes where the decrease in inside diameter is quite high the friction factor climbs to very high values. Recent boiler experience has shown that $f = .005$ is often too low. As a result $f = .006$ is in common use, although some designers still use $f = .005$ for the water side of the circuit if no steaming is present in the downcomer and if Reynold's Numbers are high.

APPENDIX "C"

Table of values of thermal conductivity, k, for a typical high carbon steel in which the carbon content is approximately 0.5%

k, thermal conductivity, BTUs per hr. ft.$^{\circ}$F

t	32°F 0°C	212°F 100°C	392°F 200°C	572°F 300°C	752°F 400°C	1112°F 600°C	1472°F 800°C
k	32	30	28	26	24	20	18

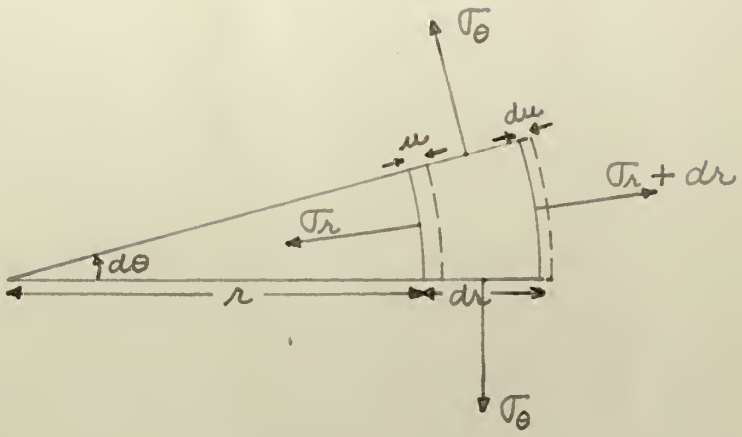
APPENDIX "D"

By the method of Case I to VII, the equilibrium circulation data for a 1.15 inch, and a 4.0 inch tube is given below at the pressures indicated.

Pressure, psi	1000		1500		2000	
Diam. O. D. inches	1.15	4.0	1.15	4.0	1.15	4.0
Diam. I. D., inches	0.95	3.48	0.89	3.34	0.85	3.18
Top dryness	0.17	0.031	0.24	0.04	0.34	0.63
Mass rate of steam pounds per hour	620	2,160	724	2,513	870	3,025
Mass rate of water pounds per hour	3,650	69,600	3,020	62,800	2,560	48,000
Density of water, lbs. per cu. ft.	46.3	46.3	42.6	42.6	38.9	38.9
Mean density in riser, lbs. per cu. ft.	20.3	39.0	21.2	35.6	20.25	32.8
Density of exit mixture lbs. per cu. ft.	10.7	28.7	11.85	29.8	12.4	27.9
Velocity of water ft. per sec.	4.44	6.35	4.55	6.74	4.65	6.22
Midpoint velocity in riser, ft. per sec.	11.79	8.29	10.48	8.2	9.6	7.45
Velocity at exit, ft. per sec.	19.15	10.24	16.4	9.65	14.6	8.68

APPENDIX "E"

THICK WALL THEORY FOR TUBES



Assume radial symmetry of structure and loading and no variation of these two along length (Z) of tube. Because of radial symmetry there is no shearing and therefore the surfaces are principle surfaces.

The radial displacement u is a function of the radius only because of the radial symmetry

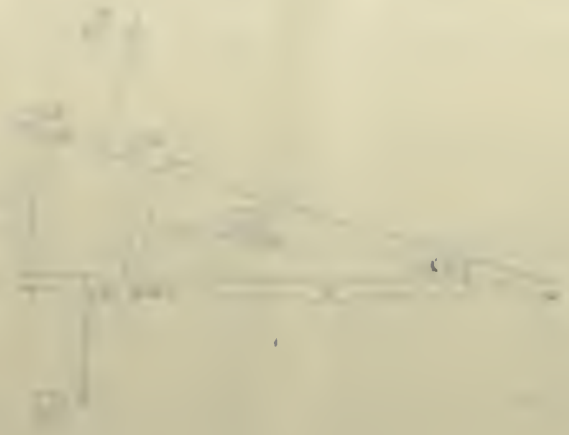
$$\text{Angular strain } e_\theta = \frac{(r+u)d\theta - r d\theta}{r d\theta} = \frac{u}{r}$$

In the radial direction the new length $= dr + u + du - u$

Therefore the strain in the radial direction is -

$$e_r = \frac{dr + du - dr}{dr} = \frac{du}{dr}$$

Equilibrium is another assumed condition in this problem and the equation is written from the right hand sketch above



$$(\sigma_r + d\sigma_r)(r+dr)d\theta - \sigma_r r d\theta - \sigma_\theta\, dr d\theta = 0$$

Divide by $dr d\theta$ and cancel where possible and the equilibrium equation is:-

$$r\frac{d\sigma_r}{dr} + \sigma_r - \sigma_\theta = 0$$

This gives three equations in three unknowns. The next step is to take Hooke's Law for a Plain Stress System and apply it to our equations as follows:

$$(1)\quad \sigma_r = \frac{E}{1-\nu^2}(e_r + \nu e_\theta) = \frac{E}{1-\nu^2}\left(\frac{du}{dr} + \nu\frac{u}{r}\right)$$

$$(2)\quad \sigma_\theta = \frac{E}{1-\nu^2}(e_\theta + \nu e_r) = \frac{E}{1-\nu^2}\left(\frac{u}{r} + \nu\frac{du}{dr}\right)$$

$$\frac{d\sigma_r}{dr} = \frac{E}{1-\nu^2}\left(\frac{d^2u}{dr^2} + \frac{\nu}{r}\frac{du}{dr} - \frac{\nu u}{r^2}\right)$$

Divide out $\frac{E}{1-\nu^2}$ and substituting in the equilibrium equation gives:

$$r\left(\frac{d^2u}{dr^2} + \frac{\nu}{r}\frac{du}{dr} - \frac{\nu u}{r}\right) + \frac{du}{dr} + \nu\frac{u}{r} - \frac{u}{r} - \nu\frac{du}{dr} = 0$$

$$\therefore\quad r\frac{d^2u}{dr^2} + \frac{du}{dr} - \frac{u}{r} = 0$$

$$\frac{d^2u}{dr^2} + \frac{1}{r}\frac{du}{dr} - \frac{u}{r^2} = 0$$

$$\frac{d}{dr}\left(\frac{du}{dr}\right) + \frac{d}{dr}\left(\frac{u}{r}\right) = 0$$

Integrate and get:-

$$\frac{du}{dr} + \frac{u}{r} = C_1$$

or

$$r\frac{du}{dr} + u = C_1 r$$

Put this in differential form and get:-

$$d(u,r) = C_1 r\,dr$$

Integrate and get:-

$$ur = \frac{C_1 r^2}{2} + C_2$$

$$\therefore\ u = C_1 r + \frac{C_2}{r}$$

Substituting in Hooke's equations (1) and (2)

(1) $$\sigma_r = \frac{E}{1-\nu^2}\left[C_1 - \frac{C_2}{r^2} + \nu\left(C_1 + \frac{C_2}{r^2}\right)\right] = \frac{E}{1-\nu^2}\left[(1+\nu)C_1 + (1-\nu)\frac{C_2}{r^2}\right]$$

(2) $$\sigma_\theta = \frac{E}{1-\nu^2}\left[C_1 + \frac{C_2}{r^2} + \nu\left(C_1 - \frac{C_2}{r^2}\right)\right] = \frac{E}{1-\nu^2}\left[(1+\nu)C_1 + (1-\nu)\frac{C_2}{r^2}\right]$$

$$-p_i = \frac{E}{1-\nu^2}\left[(1+\nu)C_1 - (1-\nu)\frac{C_2}{r^2}\right]$$

$$-p_o = \frac{E}{1-\nu^2}\left[(1+\nu)C_1 - (1-\nu)\frac{C_2}{r^2}\right]$$

Now subtract $-p_o$ from $-p_i$ and substitute for the

$$p_o - p_i = \frac{E}{1-\nu^2}\left[-(1-\nu)\frac{C_2}{a^2} + (1-\nu)\frac{C_2}{b^2}\right]$$

$$p_o - p_i = \frac{E}{1+\nu}\left[\frac{1}{b^2} - \frac{1}{a^2}\right]C_2$$

$$\therefore \quad C_2 = \frac{(p_o - p_i)(1+\nu)a^2b^2}{E(a^2-b^2)}$$

$$C_1 = \frac{1}{1+\nu}\left[-\frac{p_i(1-\nu^2)}{E} + (1-\nu)\frac{C_2}{a^2}\right]$$

$$C_1 = \frac{1-\nu}{E}\left[\frac{b^2p_o - a^2p_i}{a^2-b^2}\right]$$

Substituting these values of the constants back into (1) and (2) the normal stresses become

(1) $$\sigma_r = \frac{b^2p_o - a^2p_i}{a^2-b^2} - \frac{(p_o-p_i)a^2b^2}{r^2(a^2-b^2)}$$

Similarly

(2) $$\sigma_\theta = \frac{b^2p_o - a^2p_i}{a^2-b^2} + \frac{(p_o-p_i)a^2b^2}{r^2(a^2-b^2)}$$

In the particular case of the boiler tube the pressure is internal and p_o may be considered equal to zero. Consequently:-

$$(1)\quad \sigma_r = -\frac{a^2 p_i}{a^2-b^2} + \frac{a^2 b^2 p_i}{r^2(a^2-b^2)} = \frac{a^2 p_i}{a^2-b^2}\left(\frac{b^2}{r^2}-1\right)$$

$$(2)\quad \sigma_\theta = -\frac{a^2 p_i}{a^2-b^2} - \frac{a^2 b^2 p_i}{r^2(a^2-b^2)} = -\frac{a^2 p_i}{a^2-b^2}\left(1+\frac{b^2}{r^2}\right)$$

These equations show that σ_r is always a compressive stress and σ_θ is always a tensile stress. This tensile stress is a maximum at the internal surface of the tube where r = a

$$\therefore\ \sigma_{\theta_{MAX.}} = -\frac{p_i(a^2+b^2)}{a^2-b^2}$$

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